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LUBRICATION OF HEAVILY LOADED, LOW VELOCITY BEARINGS AND GEARS OPERATING IN AEROSPACE ENVIRONMENTAL FACILITIES

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January 1965

ARNOLD ENGINEERING DEVELOPMENT CENTER
AIR FORCE SYSTEMS COMMAND
ARNOLD AIR FORCE STATION, TENNESSEE

NOTICES

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FOREWORD

This report was prepared by the Advanced Technology Laboratories of the General Electric Company, Schenectady, New York, under Contract No. AF40(600)-1013, Program Area 850E, Project 7778, Task 777800. This research was administered by the Space Systems Office, Arnold Engineering Development Center (AEDC), AFSC, Arnold Air Force Station, Tennessee, with Mr. H. A. Person and Capt. G. Mushalko acting as project engineers.

The author wishes to acknowledge the contributions made to the program by those individuals who rendered assistance in the following areas:

J. N. Craig	- Bearing and Gear Testing
H. R. Wiseley	- Solid Film Studies and Lubricant Development
W. E. Campbell, R. J. Boot, and J. R. Gambino	- Solid Film Studies
B. Valachovic	- Radiation
T. Urbanosky	- Bearing Application Studies
P. Dean	- Gear Application Studies

In addition, acknowledgement is made to ARO, Inc., who operated the 7' Aerospace Chamber where the large scale bearing and gear tests were run. T. L. Ridings, ARO, Inc., acted as project engineer and was assisted by F. G. Moore and M. R. Hunnicutt.

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This technical report has been reviewed and approved.

GEORGE MUSHALKO Capt, USAF Technology Division DCS/Plans and Technology	DONALD D. CARLSON Colonel, USAF DCS/Plans and Technology

ABSTRACT

This report presents the results of a balanced study of bearings and gears for heavily loaded, low velocity space simulator applications, integrated with a research effort directed toward the development of materials, lubricants, application processes, and evaluation and testing techniques.

Solid film lubricants consisting of molybdenum disulfide and graphite with silicate and epoxy type binders, and thin film platings of gold and silver, were among the better performers. Molybdenum disulfide-glass, and graphite-aluminum phosphate were among the more successful solid film lubricants developed.

Apiezon L, a low vapor pressure petroleum distillate used for comparison purposes exhibited good performance characteristics.

Lubricated cylindrical, spherical, ball, and tapered rolling element bearings of 30 mm, 50 mm, and 100 mm bore sizes were tested in this effort. The gears tested were helical.

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INTRODUCTION

In the latter part of 1958, after operations in space reached a significant level, it became apparent that the existing technology of lubrication was inadequate for this environment in many respects. Of particular concern was the lack of knowledge of effective methods of lubrication in very high vacuum. Conventional fluid lubricants were soon eliminated because of severe evaporation losses, and attention was turned to dry lubrication.

Many of the dry lubrication techniques developed for earth-bound applications were heavily dependent upon low shear strength oxide films. In the absence of oxygen or other oxidant, these oxides are not replenishable in space. Once they have been removed, cold welding of virgin metal takes place and equipment malfunctions result. With the "oxide principle" of dry lubrication therefore eliminated in this new environment, direct use could not be made of much of the material knowledge accumulated in previous years.

A new start was therefore begun by government and industry to find other dry lubrication methods. This effort has been underway for approximately three years and has concentrated mainly on components for space vehicles and satellites, where the problems have been most pressing. Due to our booster limitations, these applications have inherently been characterized by light loads, low weights, small sizes, and low friction (because of vehicle-borne power limitations). Gearing had been generally neglected, with priority given to bearings of various types.

During the same period plans have been made for the larger boosters and payloads to come. These larger satellites and space vehicles will be preceded by the very large space simulation chambers now under construction. Solving the problems of bearings and gears for these simulators is therefore the next immediate objective of the technology of lubrication in space.

Problems have arisen in this area because of the change in character of dry sliding applications in simulators, compared to the types which have been under study up to now. In contrast to lightweight satellite applications, emphasis is now placed on high loads, and large sizes, with no particular size limitations on bearing and gear components. Drive power is not as severely limited as in space vehicles, so friction is not as important a criterion as formerly.

The objective of this program was to develop dependable design criteria and competitive manufacturing specifications for solid thin film lubricated bearings and gears uniquely adapted for operation in a simulated space environment. The emphasis on the lubrication effort was placed on <u>heavily loaded</u>, <u>low velocity</u> bearings and gears that are used in positioning and testing space vehicles, and other related equipment, in large space environmental test chambers such that exists at the Arnold Engineering Development Center at Tullahoma, Tennessee.

The results of the program described in this report encompassed a combined research and developmental effort directed towards meeting the requirements defined in the stated objective of the program.

SUMMARY

BEARINGS

1. Solid film lubricants considered to be among the better performers in 30 mm, 50 mm, and 100 mm bore rolling element tests consisted of:

   (a) MoS_2 + graphite + silicate binder
   (b) MoS_2 + epoxy binder
   (c) MoS_2 + glass binder
   (d) Silver

2. Solid film lubricants considered to be among the most promising ones developed in the program consisted of:

   (a) MoS_2 + glass binder (low melting)
   (b) Graphite + $AlPO_4$

The MoS_2 + glass solid film performed quite well in the 50 mm bearing tests but did not perform for as long a period in the 100 mm bearing tests.

3. Bearing configuration, and load effects, on the performance characteristics of solid film lubricants of various types were determined. For example, the endurance life of most of the solid films together with overall bearing performance appeared to favor the cylindrical and Timken tapered roller types over the more complex spherical roller bearing. The predominance of cage wear in the spherical type bearing was a case point.

4. Torque measurements indicated that solid films of MoS_2 and graphite type lubricants were less sensitive to bearing configuration, misalignment, load variations, and other factors contributing to mechanical instability, than the thin film plating type of lubricant.

5. A CBS coating (proprietary metal film) indicated considerable promise but was discarded early in the program because of prohibitive costs.

6. The performance of the solid film lubricants were optimized when all bearing elements except the rollers were coated. Not coating the rollers resulted in:

   (a) Minimizing the solid solubility attraction.
   (b) Allowed more clearance for "run-in" and redistribution of small amounts of wear debris.

7. For application of solid film lubricants to bearing surfaces, internal bearing clearances should be specified as C-3 or larger.

8. A low vapor pressure petroleum distillate grease called Apiezon L, tested for comparison purposes, performed exceedingly well in 50 mm and 100 mm bearing tests that consisted of cylindrical, spherical, and ball type bearings.

A special 1010 metal-Buna N bearing shield was designed for the retention of grease in the 100 mm ball bearing.

GEARS

1. Helical gears were used throughout all of the gear tests, i.e., 4" PD vs. 1.25" PD pinion gear. Gear lubricant combinations considered to be among the better performers consisted of:

   (a) MoS_2 + graphite + silicate against itself.

   (b) MoS_2 + epoxy against itself.

   (c) Gold against silver.

Gears were tested at loads approaching surface compressive stresses of 68,000 psi.

2. Helical gears were tested for comparison purposes with low vapor pressure greases consisting of Apiezon L, and G-300 (methyl chlorophenyl silicone grease) with a 3% silver additive. While the period of operation was considerably long, the wear and galling condition on the gear teeth indicated the lack of load carrying ability of the lubricants at the moderate load tested.

SECTION I – ANALYSIS OF SPACE SIMULATOR BEARING AND GEAR REQUIREMENTS

Prior to the lubrication and materials study an analysis of the various applications associated with vehicle moving equipment and other related space simulator bearing and gear requirements was made. It was necessary to determine the types of functions the moving surfaces perform, e.g., rolling, sliding, types of loading; e.g., radial, thrust, etc., and the specific environments encountered, before setting detailed objectives for the lubricants and materials.

A. SIMULATOR BEARING AND GEAR REQUIREMENTS

Table 1 gives a general description of space simulator applications and their associated bearing and gear requirements.

Numerous types of bearings were studied, classified into two groups where the type and specific function of the bearing was identified, and subsequently merit rated according to function as seen in Table 2, Bearing Design Selection Factors For Space Simulator Application. The design selection factors chosen are briefly described in the following subsection (IB).

B. BEARING DESIGN SELECTION FACTORS

1. Static Radial Capacity:

This is defined as the maximum radial load which the bearing can carry without suffering a permanent deformation large enough to limit the life or smooth running of the bearing. This capacity corresponds to a total permanent deformation of 0.0001 in. of the diameter of the rolling element in the bearing race and roller at the most heavily loaded contact area.

2. Thrust Capacity:

This is defined as the maximum axial load which a bearing can carry without limiting the bearing's life or smooth running ability.

3. Self-aligning Ability:

Initial and dynamic misalignment are compensated for by the bearing. Thus, full bearing capacity can be realized.

4. Space Requirements:

The geometry of a bearing type dictates its space requirements. This will probably be an important factor for several of the bearing applications.

5. Critical Angle of Inner Race:

This is defined as that angle of shaft oscillation which will allow the paths generated on the inner race by adjacent rolling elements to meet.

TABLE 1

TYPICAL SPACE SIMULATOR APPLICATIONS

Application No.	Unit	Type	No. Rqd.	Operating Pressure Torr (Note 1)	Temp. (°F) (Notes 1 & 2)	Distance to Nuclear Reactor (Ft.) (Note 3)	Exposed to Solar Simulator (Note 4)	Size	Maximum Load Thousands of Lbs.	Speed	Motion	Duty Cycle	Shaft Position	Self Aligning Required
1	Vehicle Mount Pitch Bearing (Drive End)	Radial, Equivalent to Journal Supplemented by Thrust Bearing	1	1×10^{-9}	-321	25 - 75	No	~12" Dia. Shaft	290 Transverse 12 Axial	0 to 1-2/3 RPM	Rotary Reversible	Semi-Continuous	Horizontal	Yes
2	Vehicle Mount Pitch Bearing (Floating End)	Radial with Provision for Axial Movement	1	1×10^{-9}	-321	25 - 75	No	~12" Dia. Shaft	290 Transverse	0 to 1-2/3 RPM	Rotary Reversible	Semi-Continuous	Horizontal	Yes
3	Vehicle Mount Roll Bearing (Drive End)	Radial Thrust	1	1×10^{-9} 1×10^{-9}	-321 -321	50 - 105	Yes Yes	12" Dia. Shaft	215 Transverse 430 Axial	0 to 1-2/3 RPM	Rotary Reversible	Semi-Continuous	Semi-Directional	Yes Yes
4	Vehicle Mount Roll Bearing (Floating End)	Radial with Provision for Axial Movement	1	1×10^{-9}	-321	5 - 50	Yes	~12" Dia. Shaft	215 Transverse	0 to 1-2/3 RPM	Rotary Reversible	Semi-Continuous	Semi-Directional	Yes
5	Vehicle Transfer Cart Journals	Radial with Nominal Thrust Similar to R.R. Car Journals	32	1×10^{-9} to 760	From -321 to 110 (Note 5)	15	No	6-1/2" Dia. Shaft	375 Transverse	1 RPM	Rotary Reversible	Occasional with Long Static Periods	Horizontal	Yes
6	Bearings & Gears Associated with 5-Ton Capacity Hoist Suspended From Monorail System	Varies	2 Hoists	1×10^{-9} to 760	From -321 to 110 (Note 5)	75 - 180	No	Does Not Apply			-	Occasional with Static Periods (Long)	-	-
7	Bearings & Gears Associated with 1-Ton Capacity Hoist	Varies	6 Hoists	1×10^{-9} to 760	From -321 to 110 (Note 5)	65 - 150	No	-	-	-	-	Occasional with Long Static Periods	-	-
8	Bearings & Gears Associated with 40-Ton Capacity Hoist	Varies	4 Hoists	1×10^{-9} to 760	From -321 to 110 (Note 5)	65 - 150	No	-	-	-	-	"	-	-
9	Bearings Associated with Capsule Hoist Inner Seal Door	Radial	12	1×10^{-9}	-321	60 - 145	No	1" Dia. Shaft	0.5 Transverse Nominal Thrust	1 RPM	Oscillating $\pm 90°$	"	Horizontal	Yes
10	Bearings for Personnel Entry Lock to Capsule	Radial	4	1×10^{-9}	-321	30 - 105	Yes	1" Dia. Shaft	"	"	"	"	"	"
11	Bearings & Gears for Power Unit for Vehicle Transfer Cart	Varies	1 Drive	1×10^{-9} to 760	From -321 to 110 (Note 5)	15	No	-	-	-	-	-	-	-
12	Bearings for Track Drawbridge	Radial	8	1×10^{-9} to 760	"	35 - 145	No	6" Dia. Shaft	40 Transverse	1 RPM	Oscillating $\pm 45°$	"	Horizontal	No
13	Bearings for Main Lock Door	Radial Thrust	2 1	1×10^{-9}	-30	45 - 155	No	10" Dia. Shaft	70 180	1 RPM	Oscillating $\pm 90°$	"	Vertical	Yes (for one) No
14	Bearings for Equipment Lock Door	Radial Thrust	2 1	1×10^{-9}	-30	50 - 145	No	6" Dia. Shaft	37 10	"	"	"	Vertical	"
15	Bearings for Personnel Lock	Radial Thrust	8 4	1×10^{-9}	-30	80 - 145	No	3" Dia. Shaft	1 2	1 RPM	Oscillating $\pm 45°$	"	"	"
16	Ball Bearing Screw - Jack Actuators (Misc. Applications)	-	25	1×10^{-9}	-30 to -321	15 - 180	No	1" to 6" Dia. Screw	Varies	3 RPM Max.	Oscillating Linear	"	Semi-Directional	"
17	Acme Screw & Nut Actuators (Misc. Applications)	-	10	1×10^{-9}	-30 to -321	15 - 180	No	1" to 6" Dia. Screw	Varies	3 RPM Max.	Oscillating Linear	"	"	"

NOTES:--

1. All bearings and gears may be operated under atmospheric pressure, and at atmospheric ambient temperature (760 Torr and ~70°F) during maintenance shutdowns.

2. The temperature listed is the lowest temperature which the bearing would approach under static conditions when not exposed to the radiation from the solar simulator and without some form of temperature control.

3. The nuclear reactor is assumed to be a 45 cm diameter sphere, the gamma flux at the surface of the reactor may be approximately 2×10^{13} MEV/cm^2/sec. The total neutron leakage flux may be assumed to be 1.5×10^{13} $_0n^1$/cm^2/sec at the reactor surface. The neutron flux energy distributions are:

   Fast Neutron Flux - 5×10^{12} $_0n^1$/cm^2/sec

   Epithermal Neutron Flux - " "

   Thermal Neutron Flux - " "

   The radiation level at the bearings may be approximated by use of the distances shown and the inverse square law.

4. Those bearings exposed to the solar simulator will have a heat flux imposed equal to 857 BTU/HR.FT^2 from one direction only. This exposure will be on an intermittent to continuous basis.

5. Normal operation divided between outside ambient and high vacuum conditions. The maximum temperature of 110°F is maximum outside ambient and will not be considered simultaneously with nuclear radiation and solar radiation.

6. Atmospheric constituents under the high vacuum conditions cannot be accurately predicted but are expected to consist primarily of molecular oxygen, nitrogen and hydrogen with substantial traces of ozone, molecular helium, argon and neon, and atomic oxygen, nitrogen and hydrogen.

Courtesy of AEDC

TABLE 2

BEARING DESIGN SELECTION FACTORS FOR SPACE SIMULATOR APPLICATION

"Reference" Number	Type	AFBMA Number*	Static Radial Capacity	Thrust Capacity	Self-Aligning Ability	Space Required Radial	Space Required Axial	Critical Angle of Inner Race
BEARINGS FOR RADIAL LOADS								
1	Spherical Roller	54	4	2	5	4	4	5
2	Radial Roller	37	5	0	1	4	5	5
3	Radial Roller Self-Aligning	--	5	0	5	5	5	5
4	Ball Bearing	5	1	1	2	4	1	4
5	Ball Bearing Self-Aligning	16	1	1	5	5	1	4
6	Tapered Roller	55	3	2	1	5	4	5
7	Sliding Journal	--	3	0	1	1	4	0
8	Needle Roller	47	3	0	1	1	2	1
9	Needle Roller Self-Aligning	49	3	0	5	2	2	1
BEARINGS FOR THRUST LOADS								
10	Sliding Thrust	--	0	3	1	4	1	0
11	Tapered Thrust	62	2	4	1	3	2	1
12	Cylindrical Thrust	64	0	5	1	3	3	1
13	Ball Thrust	33	0	1	1	1	1	1
14	Spherical Thrust	--	2	4	5	3	3	1

Note 1: The ratings given above represent relative merits as follows: 0 = none, 1 = very low, 2 = low, 3 = moderate, 4 = high, 5 = very high. The ratings are based on heavy duty bearing types with the same I.D.

2: *AFBMA numerical designation of bearing type

More favorable wear characteristics can be obtained if the rolling element paths overlap. Thus, a small critical angle of the bearing is desirable.

Additional information that is normally not as available as the factors previously discussed, but should be considered whenever possible are:

1. Breakaway friction torque.

2. Average running friction torque.

3. Variation of running friction torque.

4. Initial internal clearance.

5. Cost

Most of these factors were developed and the results discussed later in this report.

The results of the Table 2 study were then integrated with space simulator bearing requirements. Here, as seen in Table 3, Bearings Suitable For Space Simulator Applications, the bearing candidates that can fulfill the simulator bearing design requirements described in Table 1 are presented in a useful form.

In all, fourteen rolling element bearing types, and journal bearings were considered.

C. GEARS

An evaluation of various hand driven and electrically driven hoists indicates the availability of hoists of from 1/4 to 40 ton capacity which utilize various combinations of spur and helical gearing.

A careful study of gear efficiency and an evaluation of various types of gears in regard to their lubrication requirements leads to the following conclusions:

a) Of the generally available gear types, the most efficient is the internal gear and pinion.

b) Next in order are external spur and helical gear types and the intersecting axis bevel gear types.

c) At the lower end of the list are the worm gear types and those which have considerable sliding at the mesh point relative to the useful distance traveled.

Consider first a worm and worm gear pair. These are frequently used in industrial practice for their compactness, the relatively high ratios that can be achieved with a minimum of parts, and in certain cases for their

TABLE 3
BEARINGS SUITABLE FOR SPACE SIMULATOR APPLICATIONS

	Application No. From "Table I"	Bearing Candidates — Bearing Type "Reference" No. from Table 2: 1	2	3	4	5	6	7	8	9	10	11	12	13	14
CATEGORY I															
a) Heavy loads	1	X		X						0			0		
b) 12 in. D shaft	2			X				X		X					
c) 0 to 1 2/3 rpm	3R	X		X						X					
d) Semi-continuous duty cycle	3T												X		X
e) Rotary reversible motion	4			X				X		X					
f) Self-aligning required															
CATEGORY II															
a) Light to heavy loads	9	X		X		X									
b) Oscillating motion	10	X		X		X									
c) Occasional duty	12		X					X	X						
d) 1 rpm	13R	X		X				X	X						
	13T										X	X	X		
	14R	X		X				X	X						
	14T										X	X	X		
	15R		X		X	X									
	15T										X	X	X	X	
CATEGORY III															
a) Low Speeds	5	X		X				X							
b) Occasional duty	6		0				0	0			0	0			
c) Rotary reversible motion	7		0				0	0			0	0			
	8		0				0	0			0	0			
	11		X		X		X								
	16				X										
	17		X		X		X	0			0				

(R = Radial, T = Thrust) (0 denotes in combination) overload capability. They have two very serious drawbacks when considered for dry film lubrication, however. The first is the relatively high sliding that occurs at the teeth (hence potentially lower efficiency) and the second is the high heat dissipation requirements (again due to the lower efficiency). Industrial worm gearing has efficiency ratings that are typically from 60 percent to 85 percent as measured at the gear mesh. As normally used, the high sliding is withstood by good oil films that have great capability to heal themselves after each tooth contact in mesh. The relatively large amount of heat generated in the mesh (which damages the oil film if allowed to become too high) is carried by the oil to the sides of the gear case where it can be eliminated. The thermal rating of worm gearing is typically a practical working limit. Such gearing would seem to be a very poor candidate for a simulator application with dry film lubrication. Such a film that can withstand great sliding and which may have self-healing properties may be difficult to develop. Moreover, such a film will have virtually no ability to conduct heat from the mesh to the walls of the gear casing.

Consider next a spur or helical gear train. This type of gearing is very often found in hoists and other such equipment. It has relatively far less sliding than the worm gearing discussed above. The efficiency of spur gearing at the mesh is rarely less than 98 percent and with proper design and care in manufacture may be somewhat higher. As a result heat dissipation in spur gear and helical gear trains is not often a problem. Spur gear trains are often somewhat larger than an equivalent ratio and torque rated worm gear set. As a result of better meshing conditions, a spur or helical gear design would be a very good candidate for a dry film application.

The best possible choice of gear geometry for dry film applications is internal type gearing. Although the geometry of this type of gearing is generally less well understood by most gear designers, such gearing can have by far the highest possible efficiency at mesh. It is not difficult to achieve a design for a relatively high ratio gear set that has a mesh loss of less than one-half percent. This is due to the extremely low specific sliding of the teeth. The shapes of the contacting surfaces are very favorable to low Hertzian contact stresses.

On the basis of the above differences between types of gearing, it is probable that a specific dry film lubricant that was used in a worm and worm gear drive might show up very badly, whereas the same lubricant when used in a spur gear or an internal gear drive, would be perfectly satisfactory.

As a result, it is recommended that the gearing exposed to the simulator environment be restricted to the internal and external spur and helical gear types. If the need for data on bevel gears occurs, it can be reasonably extrapolated from the spur gear data, since meshing conditions are quite similar.

Transfer cart drives are also typically spur and/or helical gearing. Thus, this program, to enable the design of special gearing for transfer or hoist devices for simulator applications or to enable the evaluation of existing commercial equipment that might be used subject to the addition of a proper lubricating system, was limited to spur and helical gearing which should have a wide range of usefulness.

A survey of the existing drives indicates that the following size, pitch, and speed ranges are typical:

APPLICATION		TYPICAL LIMITS			
		Diameter Range	Pitch Range	Peripheral Speed Range ft/min	Type
Hand Hoist	1/4 ton	1" to 10"	6-16	up to 50	(external, internal)
Hand Hoist	4 "	1" to 10"	4-16	up to 50	"
Hand Hoist	40 "	1" to 12"	4-16	up to 50	"
Electric Hoist	5 "	1" to 12"	4-16	up to 600	"
Electric Hoist	50 "	1" to 16"	4-16	up to 600	"
Transfer Cart	electric	2" to 24"	4-10	up to 100	external

SECTION II – SOLID FILM LUBRICATION (STATE OF THE ART AND CURRENT RESEARCH PROJECTS)

A. INTRODUCTION – BASIC CONSIDERATIONS

Rapid progress in the basic understanding of solid friction has been made during the past ten years, but knowledge is still rather fragmentary regarding the mechanism of failure of solid lubricating films. Knowledge of modern friction and wear theory does offer some general guides to probable behavior of solid films and will therefore be briefly outlined.

The earliest solid friction theories were based on the concept that all surfaces, on a microscopic scale, are rough, the friction being explained as being due to interlocking of surface asperities. Modern research has confirmed the fact that the smoothest industrial sliding surfaces are so rough that flat surfaces, when brought in contact, touch at only a few points. It is no longer thought, however, that interlocking accounts for more than a negligible amount of friction. It is now believed that the friction force is largely that required to tear welded asperity peaks apart, these welds having been caused by the high contact stress on the few points in contact.

Under some circumstances, as when a very hard, rough surface slides over a softer one, the friction force due to ploughing of the hard surface can exert an appreciable influence on the total friction force. That is

$$F = \phi_1(S) + \phi_2(R) \qquad (1)$$

where F is the friction force and ϕ_1 and ϕ_2 are functions of the shear strength S and the ploughing resistance R of the softer metal.

For most practical situations the ploughing term may be neglected, and the friction force can be expressed as a linear function of the shear strength. That is

$$F = as \qquad (2)$$

where a is the real area of contact and only a very small fraction of the apparent area of contact. If we assume that the real area of contact is produced by deformation of asperities we may also write

$$W = ap \qquad (3)$$

where p is the flow pressure or hardness of the softer metal. Thus, continuing (2) and (3), we may write

$$f = \frac{F}{W} = s/p \qquad (4)$$

an equation which is very helpful in qualitative explanation of many friction phenomena.

Similar reasoning applied to the problem of wear[1], leads to the relation

$$A = K \frac{Wd}{P} \tag{5}$$

where A is the amount of metal removed from the softer metal surface, d is the sliding distance and K is a constant which is related to the probability that an asperity encounter results in the removal of a wear particle.

In considering these equations, it must be emphasized that they are not applicable except in very general terms. For example, plastics, which are commonly used as bonding agents for solid lubricants, have been given only superficial study with respect to their frictional behavior which may in many cases depart significantly from that described by equations (2) and (5).

There is also much to be learned regarding the exact mechanism of metal particle formation. It must be remembered that for many asperity encounters only elastic deformation occurs and that, of the total number in a given sliding distance, only a very small proportion results in a detached fragment. Thus much of the wear can be accounted for by surface fatigue, following repeated stressing of a given asperity[2]. Some of the particles may result from the knocking off or shaking loose of a wear fragment which has transferred to an asperity. These events are complicated by processes such as work hardening, recrystallization, oxidation, adsorption, and hydrolysis.

The local temperature rise at the asperity peaks accompanying plastic deformation and metal transfer is a very important factor in wear at high loads or surface speeds in that it influences all the complex processes which take place at asperity encounters. This is especially true of plastic bearing materials and bonding agents which are relatively ineffective at conducting away heat and tend to have low melting or softening points. Thus failure by excessive welding or seizing or by depolymerization, oxidation, carbonization, or other degenerative chemical processes tends to be aided. Stimulation by high surface temperatures of degenerative chemical reactions is also an important source of failure of inorganically bonded coatings.

Mention should also be made of the reaction of the environment, where operation is in air part of the time. The picture above, even when ploughing and interlocking are negligible so that equation (2) might be expected to hold, is greatly complicated by the fact that an oxide film is present on the surface of all base metals. This film reforms instantly on all virgin surfaces exposed during weld-breaking, transfer, and ploughing. Oxide films will always lower adhesion of metal fragment to metal and tend to have low shear strength and ductility, so that low friction is promoted and galling minimized in general. However, loose wear particle formation will tend to be promoted.

The rate of formation and the physical nature of the oxide film tend to be greatly influenced by the water content of the air. It is possible for the wear rate of a base metal to vary by a factor of several hundredfold over the relative humidity range 10 to 70 percent, with accompanying wide differences in the size and appearance of the wear particles [3] [4].

For precious metals, at temperatures above the decomposition temperature of their oxide in air, gold for example, or for base metals in vacuum, the relations described can be applied with fewer reservations.

B. PROPERTIES OF SOLID FILM LUBRICANTS

From these considerations and from experimental work which has been carried out, the physical properties for an ideal solid film lubricant can be deduced.

If the film is very thin (1-5 x 10^{-4} in.) the area of contact will be determined by the substrate metal so that

$$F = \frac{S_f}{P_m} \quad , \tag{6}$$

where S_f is the shear strength of the film and P_m is the yield pressure of the metal, and minimum friction will be realized for a thin film of a low shear strength film firmly adherent to a hard substrate. Thus Bowden and Tabor[5] haveshown that a film of the soft metal indium on steel, 2 x 10^{-4}in. thick, gives a friction coefficient as low as 0.04. This effect of substrate hardness has also been shown to hold for organically bonded films[6].

The important characteristics of an ideal solid film lubricant from a frictional point of view are therefore:

1. Low shear strength
2. High adhesion to the substrate material
3. High malleability
4. Good continuity and complete surface coverage
5. Freedom from hard, abrasive impurities

There are several other important properties which do not influence friction and wear directly, but which are of importance in choosing the solid lubricant for a given function. These are:

1. Melting point
2. Thermal stability
3. Chemical stability
4. State of particle subdivision
5. Thermal conductivity
6. Corrosion preventive ability
7. Electrical conductivity
8. Density
9. Color
10. Evaporation rate
11. Radiation resistance

Those of most interest for the present use are numbers 1, 6, 10, and 11. These will be discussed with respect to the film types discussed later.

There are two respects in which many solid lubricants are lacking. First, although conferring excellent friction lowering when present in a thin film, they have poor wear resistance, for reasons not fully understood. Second, they are not able to self-heal breaks in the film as will the film which adsorbs from a liquid lubricant of high oiliness. Even molybdenum disulfide, which is perhaps the most tenacious and wear resistant solid lubricant when applied as a powder or when rubbed into the surface, has a relatively limited life.

C. CLASSES OF SOLID FILM LUBRICANTS

There are three main classes.

1. Inorganic Compounds

    (a) Layer-lattice or laminar solids
    (b) Miscellaneous soft solids
    (c) Chemical conversion compounds

2. Solid Organic Compounds

    (a) Polymer Films
    (b) Stable high melting organic compounds

3. Metal Films

Only those classes of specific interest to the present problem, with special emphasis on progress in the past three years, are discussed here.

<u>Inorganic Compounds</u>

Layer-Lattice Solids

The materials in this class have crystal lattices in layers, the bonding between atoms in the layer being that of strong covalent or ionic forces, while those between layers are relatively weak van der Waal's forces. They can generally withstand high temperatures, and several are unusually inert. Examples of layer-lattice solids are cobalt chloride, molybdenum disulfide, graphite, tungsten disulfide, mica, boron nitride, silver sulfate, cadmium chloride, cadmium iodide, borax, lead iodide.

Because of their high melting points, high thermal stabilities in vacuum, low evaporation rates [8], good radiation resistance [9], and effective friction lowering ability molybdenum disulfide and graphite are the outstanding choices in this group.

Graphite alone is an ineffective lubricant in vacuum. However its film-forming ability can be restored by mixing with cadmium oxide or molybdenum disulfide. Most organic materials also perform this function, so that when bonded to the surface with organics it may offer effective lubricating action. Molybdenum disulfide is generally used in bonded lubricants.

Both molybdenum disulfide and graphite tend to accelerate corrosion as rubbed films, the former by hydrolysis which produces an acid and the latter by galvanic action.

Although many layer-lattice solids are effective lubricants, this structure in itself is not a guarantee of good lubricating ability. Examples of layer-lattice solids which are ineffective lubricants are mica and boron nitride. In each case adherence to the metal surfaces is very poor. This is at least one reason for the ineffectiveness of these lubricants.

Chemical Conversion Coatings

These coatings are inorganic compounds developed on the surface by chemical or electrochemical action. Although many of these films are not strictly solid lubricants, they are very effective in preventing wear. One of the best known treatments for steel, phosphating, which coats the surface with a layer of mixed zinc, iron, and manganese phosphates, has been shown to increase the life of organically bonded coatings by a large factor for operation in air. Its primary purpose seems to be to provide a reservoir for lubricant. It is not effective in high vacuum, perhaps because of loss of retained water. Other films in this class are sulfide, chloride, fluoride, oxide and oxalate films.

Sulfide films can be formed on steel surfaces by immersion in molten mixtures of inorganic sulphur-containing salts such as sodium thiosulfate, sodium sulfide and potassium ferrocyanide. These treatments sulfide the surface and produce case hardening as well. A large variety of treatments is described in a series of Russian translations abstracted in reference 10. A treatment of this type, developed in France, is available in this country. It has been extensively studied by the Hamilton-Standard Propeller Division of United Aircraft Corporation. These treatments offer an interesting possibility for use as a base for bonded solid lubricants.

A number of different hardening treatments as well as flame spraying of tungsten and titanium carbides provide excellent wear resistance, although the friction coefficient may not be as low as desired. Some of these may also provide good bases for low shear strength films. Several of these treatments have been recently shown to be very effective in a Falex test in chlorinated silicone diester and SAE 10 mineral oils [(11)].

<u>Solid Organic Compounds</u>

Polymer Films

Although polymers have been widely used when purified with fillers to provide added strength, the only materials which have received wide use are polytetrafluoroethylene and polychlorofluoroethylene. Both these materials provide very low friction and excellent wear resistance at moderate loads and speeds as thin films, but tend to gall and wear at high loads or speeds. Both materials are difficult to apply in an adherent form and have an unduly high coefficient of expansion.

All polymer materials possess low thermal conductivity, so that frictional temperatures reach higher values at high loads and speeds than for metals or ceramics. The film then fails by melting, softening, or chemical degradation.

Although early work indicates that with the exception of PTFE, the modern theory of friction is applicable to polymers in bulk [12], more recent studies indicate that for thermoplastic materials friction is less dependent on plastic yield than for metals [13], so that

$$F = KW^n \tag{7}$$

where n lies between 0.67 and 1.0.

Results for reinforced thermosetting plastics, which would be expected to approximate the behavior of bonded-solid lubricants, also show a slight decrease in friction coefficient with load. These departures do not appear to be sufficiently marked to prevent qualitative application of equations (2) and (6) to thin films.

Some of the non-polar type thermosetting resins such as high density polyethylene and polypropylene can be compounded to give relatively high melting points and have been shown to be useful as antiwelding agents in sheet form in deep drawing operations, but they lack sufficient adherence to provide lasting lubricant films. Perhaps the recent finding that a suitable organic, chemisorbed, oriented monolayer laid down on the metal surface greatly increases adherence of non-polar polymers can be used to improve their effectiveness. For example, a Langmuir-Blodgett type stearic acid laid down so that the hydro carbon chain, soluble in polyethylene, was oriented away from the surface, doubled the adhesive bond strength of polyethylene [14].

All in all the possibilities for polymer films do not appear to be very promising for simulator use because of the high loads involved. Their most promising use would seem to be as binders for inorganic solids, which are discussed later.

Stable High Melting Organic Compounds

A tremendous amount of work has gone into developing liquid lubricants of high thermal stability, but relatively little has gone into the development of stable organic solids, although at least two, both pigments, have shown promise as thickeners for high temperature greases -- copper phthalocyanine and phenanthrene. As far as can be determined, neither material has been used as a bonded solid lubricant, but phthalocyanines, which are organo-metallic chelates, have been used as solid film lubricants by reacting the metal free phthalocyanine with the metal surface at high temperature (15). Thus the strong chelate bond gives good adhesion and the laminar, easily sheared compound provides lubricant action. Phthalocyanine bonded in this way, has been shown to give very low friction at 1000° F, and its mixtures with MoS_2 have been used successfully as powders applied to the surface of rolling contact bearings in a gas stream(16), but the bonded film is not long-lived and means must still be found to renew it continuously.

All the work to date has been carried out at 600-1100° F where strong reaction takes place between metal-free phthalocyanines and the metal surface. At the lower ambient temperatures of the subject operation, there is some question whether the metal-free material will react with the metal surface. If it does not, it will probably not lubricate any more effectively than the metallized phthalocyanines, which are relatively poor lubricants.

Metal Films

The physical properties of soft metals come close in many respects to those outlined in "Properties of Solid Film Lubricants" for a good solid lubricant. They have low shear strength, can be bonded strongly to the substrate metal as continuous films, have good lubricity, and high thermal conductivity. Some of the more promising metals are listed in Table 4 below (17).

TABLE 4

PROPERTIES OF SOFT METALS

Metal	MhO Hardness	Melting Point °F
Gallium	-	85
Indium	1	311
Thallium	1.2	579
Lead	1.5	622
Tin	1.8	450
Gold	2.5	1945
Silver	2.5 - 3	1762

The melting points of Ga, In, and Sn are too low for situations where high surface temperatures are developed -- high speeds and loads. Pb and Tl are borderline in this respect, while Au and Ag are quite satisfactory. All but Ga can be applied to give satisfactory corrosion protection, all have low evaporation rates to 200° F, and should not give trouble with mild radiation dosage.

Ga is a special case, since it is above its melting point under most conditions of operation. It is also too reactive with many metals to be suitable, but with other metals such as 440C stainless steel, and with ceramics such as boron carbide or aluminum oxide, which can be applied as undercoats, it has been shown to be an effective lubricant in high vacuum (18). It is possible that, because of its excellent wetting characteristics for metals and ceramics, it could be used as base for lubricating "muds" containing solid lubricants such as MoS_2 or graphite.

Base metals all form oxides in air. On soft base metals, such as In and Pb, the oxide is probably harder than the substrate, and its formation and removal during slide is probably an important factor in reducing the wear life of soft, base-metal films in air operation. In vacuum (10^{-6} torr), oxide formation will probably not influence wear too greatly and the effectiveness of the soft metal film should be increased. In fact, barium, a soft metal which would be impossible to use in air because of its rapid oxidation, has shown excellent lubricating properties in x-ray tubes [19].

Gold and silver have both shown promise in a number of lubricating uses in air and in vacuum. They are especially useful at high temperatures in air, no doubt because they do not form oxides and are much softer under these conditions [17, 18, 20, 21, 22, 23].

Under severe conditions and at high temperatures these films may fail by oxidation of the substrate base metal through pores in the thin noble metal followed by eventual spalling-off of the film. In any event, the substrate metal will be exposed periodically during the sliding process by ploughing through or adhesional transfer of the noble metal film. When this process takes place in air, the substrate metal will reoxidize and will thus tend to prevent good readhesion of transferred noble metal at the exposed points. Repetition of this sequence will in time promote the development of wear particles. It is believed by W. E. Campbell[7] that great improvement in film life may be achieved by interposing between the soft metal film and the substrate base metal a harder noble metal film such as rhodium or platinum-iridium. For maximum adhesion of the soft noble metal, the metal of the intermediate film should alloy both with the substrate metal and the soft noble metal lubricating film. In some cases this may require more than one intermediate bonding coat. For example, silver does not alloy to steel so will tend to lack adhesion. A flash of hard nickel will bond well to the steel, but this tends to oxidize and should be coated with the hard metal rhodium. The silver of thickness .00005" to .0002" would then be coated on the outside. In addition to promoting better adhesion and preventing loose particle formation, this triplex film will provide greatly increased corrosion protection. The soft noble metals are amazingly long-lived under such conditions. In some experiments on sliprings sliding against palladium-alloy brushes at moderate Hertz stresses [24], a copper slipring, coated with .000005" gold over .000002" rhodium, had run 10,000 hours continuously when last observed and there still appeared to be complete coverage of gold in the wear track. The wear scar on the brush also appeared to be coated with gold. In this case the gold apparently transferred back and forth continuously from brush to slipring surface without loose particle formation. This behavior is predictable from energy considerations. It has been postulated that a loose wear particle will not form when the energy of adhesion of film to substrate is greater than its residual strain energy[25]. At higher stresses particle formation may occur by surface fatigue, but sandwich type films like this merit careful consideration.

D. BONDED SOLID FILM LUBRICANTS

Many soft solids, when rubbed into a surface, appear to lack the ability to form an adherent, continuous film. Mica and boron nitride are especially lacking in this regard and even graphite is poor in its film-forming under some conditions. Even when film-forming is good as with MoS_2, the film is very thin. Since solid films suffer attrition, as with most other solids, by loose particle formation, it is either necessary to prevent particle formation or to provide a thicker film. Binders for solid lubricants function by improving adhesion and thus film-forming and by enabling a thicker film to be applied to the surface. They may also aid in slowing down particle formation, but this process is too little understood at present for any clear cut criteria to be applicable.

Many variables are involved in the functioning of bonded solid lubricants. They are:

1. Those dependent on the composition and properties of the coating.

   Type of binder
   Binder-solid lubricant ratio
   Type of solid lubricant
   Film thickness

2. Those dependent on the properties of the substrate.

   Surface cleanliness
   Surface roughness
   Surface hardness
   Type of surface film

3. Those dependent on operating conditions.

   Load
   Speed
   Temperature
   Environmental pressure variation
   Environmental contamination
   Type of motion

1. Properties of the Coating

(a) Type of Binder

Although a large variety of binders has been used, the selection has been purely empirical. There appears to be no study in the literature offering basic guides to the selection of a binder.

Among the organic coatings, the thermosetting resins have been used almost exclusively, especially when severe operation is expected. Baked coatings appear to give longer life than unbaked coatings, the baking procedure depending upon the resin type employed. Vinyl, acrylic, alkyd, polyurethane, silicone, phenolic, and epoxy resins have all been used. From a study of the successful coatings on the market it would appear that the most promising types for the subject use lie in the epoxy, phenolic and alkyd classes.

In general, the bulk frictional properties of the thermoplastic resins would seem to be more suitable than those of the thermosetting resins for low friction coatings. However, they are low in adhesiveness and have low melting points. Possibly graft polymerization to confer adhesive groups would produce more suitable binders [26, 27]. Surface treatment to improve adhesiveness has already been suggested [14]. Modified vinyl resins are used for air-drying coatings. None of the coatings in this class are, of course, suitable for high temperature use.

In many industrial and military uses, oil and solvent resistance is important and tests for these properties in military specifications eliminate some very effective coatings. Oil and solvent resistance do not appear to be important for the subject use.

Several effective inorganic binders and solid lubricant coatings have been developed in recent years, most of them for use in the range 500° to 1500° F, and at high rubbing speeds. Almost all of these are ineffective at low temperatures and speeds, probably because the binder is too brittle under these conditions. Only those which may be usable in the present program will be described here.

Comprehensive tests at the Naval Air Material Center, Philadelphia, Pennsylvania, using combinations of MoS_2, graphite, and sodium silicate (1 Na_2O; 2.90 SiO_2 by weight) have been made [28]. The screening was done using a Falex tester in air at 77°F with 1000 lbs. gauge load. Selected compositions were run in a ball bearing life test on 204 bearings in air at 330° F at 1250 to 10,000 rpm. The bearings were loaded 5 lbs. in thrust and 3 lbs. radially. A solid to binder ratio of 78:22 was used in most of the tests, which included a few with potassium silicate, sodium metaborate, and sodium metaphosphate. A mixture of MoS_2 and graphite containing 9 percent graphite was found to give optimum results with the sodium silicate binder. This mixture, 71 parts MoS_2, 7 parts graphite, and 22 parts sodium silicate, was coated on the retainers and races of the 204 bearing and performed satisfactorily at 10,000 rpm. in vacuum to temperatures to 750° F. This coating has also beeen tested in vacuum by several other laboratories at temperatures from 77° to 400°F in vacua or 10^{-6} to 5×10^{-8} torr [23, 29, 30]. It is stable to moderate radiation dosage. It is available from all leading solid film lubricant manufacturers.

PbI, PbI_2, PbS, PbO, Z_2Cl, and ClI_2 were also given preliminary trial with the sodium silicate binder. Chemical reactions took place with some combinations of solid lubricant and binder, but several compatible combinations were found which merit additional trial.

The same authors have recently completed studies of a number of additional binders and solids using the Falex test at 1000 lbs. gauge load in air at 770° F [31]. For the binder studies, the 10 percent graphite in MoS_2 mixture was used for the solid lubricant in the same ratio as for the silicate. Binders were sodium metaborate, beryllium fluoride, sodium hexametaphosphate, sodium and potassium silicate, phenolic, epoxy, and silicone resins. The beryllium fluoride bonded coatings gave the best wear life and sodium metaborate, potassium silicate, and sodium hexametaphosphate all compared favorably with sodium silicate. All these coatings behaved better than the organically bonded coatings. Bearing tests were made on the standard coating using the standard coating and M-10 balls, retainers and races in air at 77° F and 750° F and in vacuum of 10^{-5} to 5×10^{-6} torr at 1000° F and gave satisfactory performance.

It should be noted that the high-load runs were only made in air. The vacuum runs were generally made at temperatures above 500° F and at high speeds and low to moderate loads. It by no means follows that the sodium silicate bonded coating will show the same superiority in high vacuum at high loads and moderate speeds, and low temperatures. Effective coatings at high rubbing speeds and high temperatures, such as the $PbO:SO_2$ coatings developed at National Aeronautics and Space Administration [32] have been shown to be poor at high loads, low speeds and temperatures. On the other hand, only one solid lubricant, MoS_2 with 10 percent graphite, was tested for all binders, and only the sodium silicate composition was tested in vacuum. This leaves a wide range of possibilities for additional tests.

It should also be noted that all the binders tested are soluble in water. Although the baked silicate coat is resistant to water in laboratory tests, prolonged exposure to high humidity may reduce its effectiveness.

A large amount of work on solids and inorganic binders has also been carried out by NASA, Cleveland, Ohio, and by the Midwest Research Institute, Kansas City, Missouri. The NASA work has been aimed almost exclusively at high surface speeds and temperatures. The MRI work has also been concerned primarily with extending the temperature range of solid lubricants, but some of the compositions selected have been tested in room temperature and in high vacuum. One of the best compositions selected consists of 4 PbS, 8 MoS_2, 1 B_2O_3 by weight applied from a slurry in 70 percent ethyl alcohol and baked at 1000°F - 1500°F in a nitrogen atmosphere [33]. The purpose of the PbS is to improve wear resistance at temperatures above 500° F. It tends to reduce effectiveness at moderate temperatures. $MoS_2:B_2O_3$ compositions in the range 10-15 to 1 should be even more wear resistant at temperatures from -100° to 200° F. Coatings of MoS_2 bonded with trisodium phosphate and sodium tetraborate in the ratio 4 MoS_2 to 1 binder by weight show good performance in air at room temperature [34].

(b) Binder Solid Lubricant Ratio

In general the bonding agents having high adhesiveness have poor wear and frictional characteristics. It is necessary to find an optimum ratio of solid to binder for each solid and resin type. With the commonly used MoS_2-graphite solid and organic resin binders the ratio runs in the neighborhood of 2:1, with inorganic bonding agents it runs from 4:1 to as high as 20:1 when high temperatures are involved.

(c) Type of Solid Lubricant

MoS_2 appears to be the outstanding choice for moderate temperatures and high loads. Many compositions include graphite in the mix in the range 5-15 percent. The graphite appears to exercise a synergistic effect, but no explanation for this has been given. Graphite bonded with organic resins should perform well, but is seldom used, although there is little published evidence that it is not effective. It has been shown to be ineffective with inorganic binders [28]. Soft metals have been included in some compositions, but it is doubtful whether these offer any advantage for the subject use. Teflon compositions in general have not been effective at high loads.

(d) Film Thickness

It is generally believed that the optimum thickness for resin-bonded films lies in the thickness range .0002 - .0004". Recent very careful work using two methods shows that wear life increases for film thicknesses from .0001" to .0003" [35]. Above .0003" there is no advantage in additional thickness. Coatings of thickness from .0001" to .001" were used. For inorganic coatings the optimum thickness will probably be different. From unpublished work, it may be higher.

2. <u>Properties of the Substrate</u>

It is clear from equation (6) that the hardness of the substrate would be expected to influence friction. Several investigators have shown this to be true, the coefficient of friction being lower and the wear life being greater for a given coating for a harder alloy of a given metal. There are other specific effects, not fully explained, which are independent of the hardness. For example, it has been shown in Falex tests in air at 77° F, that a MoS_2- 10 percent graphite, bonded by eight different materials, including three organic resins on AISI-C1137 steel, vary anywhere from two to more than ten times the wear life running against a molybdenum pin, than against a AISI 3135 steel pin of approximately equal hardness. Also, a number of metal sulfides containing 10 percent graphite and bonded with sodium silicate on a C1137 steel give no wear protection running against 3135 steel, Inconel X and Ti-6AI-4V pins, but give protection when running against molybdenum (good) or a molybdenum-containing steel (M-10, fair). Similar improvements in behavior were found with 204 bearings at 10,000 rpm in air at 750° F when M-10 tool steel and molybdenum were used instead of Inconel X or AISI-TI steel for the balls, races, and retainers, using the silicate bonded MoS_2-graphite coating. Large differences in wear life in modified-Timken machine tests have been found for $PbS:MoS_2:B_2O_3$ coatings on a number of substrate metals (34). No explanation is offered for these effects. At least two possibilities can be seen readily. The first is that the oxide on the different metals will react with some binders and solids and not with others, leading to a variety of results. Further, the constituents of the solid lubricant, especially the S, may react with the metal oxide or the metal. In the case of molybdenum this could have the effect of continuous renewal of a MoS_2 film at the rubbing interface.

Where film compatibility and wear life is satisfactory, it is obviously an advantage to use a wear-resistant surface under the bonded lubricant. A molybdenum coating would appear to be worth applying when bonded MoS_2 films are used. Hard metal coatings should also be considered. A high cobalt alloy gave good wear characteristics in ball bearing tests in high vacuum (18).

Freedom of the surface from contaminating organic films or from oxide films has been thought to give the best adhesion. This is undoubtedly necessary to insure uniform, adherent, and continuous chemical conversion films upon which to apply the bonded lubricant. However, as has been mentioned, some adsorbed films may improve adhesion in certain cases, and where an oxide film enters into the interfacial reaction to produce favorable effects, it may be desirable to precoat with this oxide.

Phosphate undercoatings on steel have been found to improve wear life of bonded coatings tremendously, presumably by providing a porous surface film to hold reserve lubricant. There is some question as to whether these are desirable in vacuum because loss of water held in the coating may effect its adhesion. The safest treatment is probably a vapor blast of all surfaces to be coated. A surface roughness of 15-25 microinches has been shown to give the best wear life with resin-bonded MoS_2-graphite, in modified Timkin tests (36).

Prolonged exposure to high humidity ($<$ 70 percent r. h.) with operation can effect wear life of bonded coatings, reducing it with some and increasing it with others. Another problem at high humidity is corrosion. Improvement may be obtainable with a suitable subcoating surface conversion treatment or by inclusion of inhibitors in the coating, but little progress has yet been made.

3. Operating Variables

The wear life varies with the operating conditions imposed by space simulator applications, and these conditions are most generally quite severe. The moderate temperature range, fortunately, provides some amelioration of conditions thereby allowing consideration of organic binders in addition to the inorganic ones.

E. SOLID LUBRICANTS AS ADDITIVES

Although oxidation and evaporation loss of organic fluid lubricants are problems at very high vacua where long exposures at high temperatures are needed, in the present case, relatively long life should be possible [21, 37]. In any case, relubrication should presumably be possible. By using surface treatment to prevent oil creepage and design to minimize evaporation, organic lubricants of moderate oxidation and radiation stability can be expected to last at least a year. This means that low-temperature lubricants of minimum vapor pressure could conceivably be selected as bases from the petroleum, silicone, and polyester classes. If this reasoning is correct, greases containing small percentages of MoS_2 or graphite would be interesting possibilities. By suitable thickening (4-5 percent soap), and additives, it is possible that excellent wear prevention with greater corrosion resistance could be achieved than can be provided by solid film lubricants alone.

F. GENERAL DESIGN CONSIDERATIONS

The limitations of solid film lubricant coatings must be kept in mind in designing bearings and other sliding parts. These are that the film is not self-renewing and that its wear life is generally limited. It is obviously desirable that coatings be provided which would last beyond the expected life of the bearings with every precaution being taken to maximize the life under operating conditions. Based on the present state of the art, however, it is more realistic to take the approach of establishing the wear life limits that various types of solid film lubricants will incur, thereby making use of these lubricants within their operating limitations rather than ruling them out because of not meeting unrealistic goals such as life of the bearing per se.

All rubbing surfaces should be coated, if possible. That is, shaft and bearing surface for a journal bearing, and race, retainer, and balls or roller surfaces for a rolling-contact bearing. Available clearances and solubility considerations will affect this decision.

Journal bearing edges should be rounded and clearances should be sufficient to minimize scraping-off of the coating during assembly, as well as providing for escape of wear debris during operation. Abrasive contamination should be minimized in storage and assembly. During use in air, abrasive and other contamination should be minimized by suitable shielding. Escape of oil vapors in vacuum may also be slowed by shields or seals.

When feasible, and where possible, provision may be made in design for renewal of coating. Three such approaches applicable to the present simulator problem have been used. The first approach is "stick lubrication", in which a compact of solid lubricant is applied continuously to one or both moving surfaces, probably better adaptable to sleeve bearings. A second approach is to provide slots or pockets in the bearing surfaces, i.e., retainer element, etc., which would contain compacted lubricant powder or a bonded lubricant of high solid to binder ratio, from which solid would be smeared over the wear track [38, 39]. A third approach is to make one of the bearing elements, such as the retainer, out of a composite of metal and solid lubricant, the metal to lend structural integrity.

These methods, like that of applying a thin film directly, have their own particular problems and limitations. Several pertinent ones are:

1. Means of removing the continuous generation of wear debris must be provided in order to prevent jamming of the bearing.

2. Structural integrity of a metal + lubricant composite can often be a problem.

3. Cost of item 2 fór large size rolling element bearings may be a prohibitive factor.

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SECTION III – LUBRICATION STUDY AND DEVELOPMENT

This section concentrates on specific areas of inorganic and organic materials and processes as a potential means of developing solid film lubricants. Some of it is theoretical; therefore, the extent of its usefulness has yet to be proven. Other parts however consist of theory, carried further to the development stage at General Electric's Advanced Technology Laboratories (ATL), and subsequently tested in small scale and large scale bearing operation reported in Sections VIII, IX, and X.

The first part of the lubrication study deals primarily with inorganic lamellar compounds. The second part consists of G.E.'s developmental effort with a number of potential solid film lubricants and the subsequent development of a low sintering temperature MoS_2 - glass lubricant. The third part discusses the use of Chelates as solid film lubricants followed by a brief consideration of plastics as solid lubricants among the organic family.

A. INORGANIC LUBRICANT STUDY

The usefulness of the two classes of solid film lubricants, organic and inorganic, has been demonstrated. Organic lubricants are more sensitive to radiation damage than inorganics and in addition are subject to outgassing at the low ambient pressures of interest.

A large majority of inorganic compounds have structures made up of infinite three-dimensional complexes. Many of these compounds have been observed to provide lubrication in thin film applications. It is generally accepted, however, that compounds, i.e., containing two dimensional complexes with weak interlayer bonding, having a lamellar structure are more effective lubricants. Three classes of lamellar compounds can be distinguished by the type of bonding, van der Waals, hydrogen, ionic, holding the layers together. Ionic bonding is found in chlorite minerals, micas and some clays. Hydrogen bonding, between layers, is reported in aluminum hydroxide. For reasons to be discussed later, these materials do not appear useful as lubricants in the current application. Most lamellar compounds are those in which layers are bonded by van der Waals forces. Since these appear to be the most promising class of lubricants, a tabulation by anion type is given in Table 5.

The primary criteria for the selection of specific lubricants for evaluation in vacuum environments should be based on the intrinsic lubricating properties. A secondary consideration is the anticipated effect of the cryogenic conditions and vacuum on thin film lubricant behavior. The reported lubricating properties of these materials should represent the best guidance in selecting materials. Unfortunately, not many of these materials have been evaluated under vacuum conditions and the data does not lend itself to generalization. Information on lubricating properties at atmospheric pressures is more plentiful but the results are not very meaningful in predicting behavior in vacuum environments. A well-known example is the degrading influence of vacuum on graphite lubrication. Absorbed vapors which are apparently responsible for lubricating properties of graphite are not present in vacuum and hence its performance degrades. In order to illustrate that converse behavior is possible, hypothetical examples will be used. Boron nitride appears to be a poor lubricant in atmospheric evaluations. If this behavior is caused by oxide layers on the powder surface (commercially available BN has an appreciable oxygen content) or oxide films formed during service, careful preparation and handling of the powder could result in markedly improved lubrication in a vacuum environment where sources of oxygen contamination are absent. Halides can be expected to perform similarly. These materials hydrolyse so readily that exposure to minute partial pressure of water vapor could cause a variety of hydrolysis products to form on the surfaces. It is reasonable to expect these products would have an adverse effect on lubrication behavior and their absence in vacuum would lead to improvement. A generalization can be made that such classes of materials can be expected to be better lubricants* in

*Vacuum operation also allows certain treatments to be made which are not feasible otherwise. For instance the surface of BN powders could be modified by sulfiding reactions to form boron sulfides without the danger of hydrolysis.

TABLE 5

LAMELLAR COMPOUNDS

CHALCOGENIDES			HALIDES			OXIDES	NITRIDES	OTHERS
SULFIDES	SELENIDES	TELLURIDES	CHLORIDES	BROMIDES	IODIDES			
MoS_2	$MoSe_2$	$MoTe_2$	$CdCl_2$ (2)	$CdBr_2$(1)	CdI_2 (1)	MoO_3	BN	Graphite
WS_2	WSe_2	WTe_2	$TiCl_2$ (1)	$FeBr_2$(1)	CaI_2 (1)	$Mg(OH)_2$ (1)		
TiS_2 (1)			$FeCl_2$ (2)	$CoBr_2$(1)	MgI_2 (1)	$Cd(OH)_2$ (1)		
ZrS_2 (1)			$CoCl_2$ (2)	$NiBr_2$(1)	PbI_2 (1)	$Ni(OH)_2$ (1)		
SnS_2 (1)			$NiCl_2$ (2)		MnI_2 (1)	$Co(OH)_2$ (1)		
PtS_2 (1)			$MgCl_2$ (2)		FeI_2 (1)	$Fe(OH)_2$ (1)		
			$ZnCl_2$ (2)		CoI_2 (1)			
			$MnCl_2$ (2)		TiI_2 (1)			

(1) CdI_2 Structure

(2) $CdCl_2$ Structure

vacuum than in normal atmospheres. Lubrication in vacuum should be a more reliable measure of intrinsic properties. It is apparent that behavior in atmospheric pressures is an uncertain guide to behavior in vacuum. In some cases, however, atmospheric performance can be informative. For instance, there is no reason to expect that the poor lubrication displayed by mica would be improved in vacuum.

The physical and chemical behavior of the lubricant in a spatial environment will also determine the usefulness of specific lamellar compounds. Material losses by evaporation or decomposition are undesirable since both film thickness and composition affect lubrication. Contamination of the vacuum system should be kept at a minimum. Some of the halides listed in Table 5 have appreciable vapor pressures at room temperature. Vapor pressures of some representative lamellar compounds at 300 K are shown in Table 6. In order to eliminate materials on the basis of their vapor pressure, a maximum allowable rate of vaporization must be established. An arbitrary maximum rate would be to assume that evaporation rate cannot exceed the rate which would cause all of the material in the film* to be removed in the service time specified.** This value for CdI_2 is about 5×10^{-10} gm-cm^{-2} - sec^{-1}.

The rate of evaporation is related to vapor pressure and can be calculated using the equation:

$$\mu = P \left(\frac{M}{2\pi RT} \right)^{1/2} \tag{1}$$

where P = vapor pressure
M = Molecular Weight
R = Gas Constant
T = Temperature
μ = Rate of evaporation gms/cm^2/sec

The calculated rate, 6×10^{-15} gm- cm^{-2} - sec^{-1}, for CdI_2 which has the highest vapor pressure of the materials listed in Table 6 is much lower than the maximum rate specified above. It is doubtful that simple evaporative losses can be the basis for excluding from consideration any materials listed in Table 5.

Since any decomposition products on the surface can be expected to be detrimental, it will be assumed that decomposition cannot be tolerated. The equilibrium composition can be calculated from thermodynamic data when available. The free energies of dissociation reactions of the type:

$$MX_{2(s)} = M_{(s)} + X_{2(g)} \tag{2}$$

can be calculated for a number of amellar halides. The stability of these compounds is indicated by the low chlorine pressures, 10^{-60} atmospheres, necessary for dissociation of cadmium chloride. Similar data on other classes of lamellar compounds are not readily available. The relative stability of these compounds can be approximated from heats of formation. The notable trend is the decreasing stability

* about one mil
** about one year

TABLE 6

VAPOR PRESSURE OF SELECTED LAMELLAR COMPOUNDS AT 300K(*)

COMPOUNDS	P, mm Hg
$CdCl_2$	1×10^{-20}
$CdBr_2$	3×10^{-16}
CdI_2	3×10^{-15}
$CoCl_2$	2×10^{-30}
$FeCl_2$	8×10^{-19}
$MgCl_2$	8×10^{-24}
$MgBr_2$	2×10^{-23}
MgI_2	5×10^{-25}
$MnCl_2$	4×10^{-23}
MoO_3	1×10^{-30}
$NiCl_2$	1×10^{-29}
$TiCl_2$	1×10^{-36}
$ZnCl_2$	4×10^{-16}

* O.Kubaschewski and E. Evans, Metallurgical Thermochemistry, John Wiley & Sons (1956).

with increasing anion atomic number in the halide compounds; i.e. iodides are the least stable compounds. Dissociation of the iodides to the elements is unlikely, however, as evidenced by the high free energy of the reaction:

$$Cd_{(s)} + I_{(g)} = CdI_{2_{(s)}} \quad -\Delta F_{298}, \text{ 37 Kcal} \tag{3}$$

Of course, the exact mode of dissociation is unknown. Disproportionation by reaction of the type:

$$2\ CdI_{2_{(s)}} = 2\ CdI_{(s)} + I_{2_{(g)}} \tag{4}$$

is one possibility. The fact that monoiodides of the metals of interest have not been reported[3] (ZnI is an exception), strongly suggests that iodide reactions such as (4) are not significant. Not enough data is available to comment on disproportionation reactions such as:

$$2\ MX_2 = MX_4 + M \tag{5}$$

which have been observed at high temperatures. Only one class of compounds can be definitely eliminated because of instability. $Mg(OH)_2$ is undoubtedly the most stable of the hydroxides of interest. The equilibrium water vapor pressure for the reaction:

$$Mg(OH)_{2_{(s)}} = MgO_{(s)}\ H_2O_{(g)} \tag{6}$$

is only 10^{-8} atmospheres at 300K. The formation of MgO would therefore be highly likely in a spatial environment.

Lubricants can be applied with or without binders. Generally, lubricants applied without a binder have lower coefficients of friction but have shorter effective lifetimes than bonded films. Compatibility with binders is another criteria which can be used in evaluating lubricants. The reactivity of the halides should limit the choice of suitable binders for these materials. Halide-substrate reactions are also likely but it is not certain the results would be necessarily detrimental. Such reactions might be useful in developing a bond in lubricating films. Corrosion effects by the sulfides, oxides, nitrides and graphite should be negligible.

B. INORGANIC LUBRICANT DEVELOPMENT

Thin, solid lubricant films can be applied with or without binders. A pure lubricant film generally has a slightly lower coefficient of friction, but the effective lifetime of the film is short as compared with the lifetime of a properly bonded film. The present application demands effective lubrication over extended periods of time without maintenance. Pure lubricant films could not be expected to meet this requirement. Consequently, a study of various binder-lubricant combinations was initiated in a search for materials providing optimum lubrication under the expected environmental conditions; (i.e., operation in dry or humid air at temperatures to 110°F, and in hard vacuum with temperatures ranging from 110°F to -300°F).

Various investigations reported in the literature indicate that graphite (C), and molybdenum disulfide (MoS_2), are among the best of the solid lubricants. Graphite is known to lose lubricity in vacuum when the adsorbed water film evaporates. Molybdenum disulfide exhibits its best lubricating properties in vacuum but deteriorates to some degree in the presence of normally humid air. Tungsten diselinide (WSe_2), which closely resembles MoS_2 in many ways, is reported to be somewhat more stable than MoS_2 at high temperature. No report of its behavior under humid conditions has been found but, according to P. H. Bowen (AEDC-TDR-62-51), the quantity of adsorbed or contaminating gas evolved from powdered WSe_2 in vacuum is much less than found with powdered MoS_2.

In bonded solid lubricant films, the bond itself may be considered as providing the atmosphere immediately surrounding the bulk of the lubricant particles. Thus it may be possible to use graphite in vacuum if the bonding agent is capable of supplying the small quantity of water vapor that is required for optimum lubricity. Conversely, MoS_2 may retain its very low coefficient of friction when exposed to high humidity if the bond is initially moisture-free and forms an impervious protective film on the solid lubricant particles.

The following brief list of representative materials were chosen to explore these possibilities:

1. Moisture-containing materials for bonding graphite powder
    a. Non-crystalline
        1. Water soluble aluminum triphosphate "glass"
        2. Water soluble sodium silicate "glass"
        3. Non-soluble silica gel
            (a) hydrated ethyl silicate
            (b) colloidal silica (Nalco)
    b. Hydrated crystals
        1. Low melting barium hydroxide
        2. Water soluble alum
        3. Colloidal suspension of needle-shaped aluminum hydroxide (Baymal)

2. Moisture-free materials for bonding MoS_2 powder
    a. Non-crystalline
        Low melting sulfur-type glasses
    b. Micro-crystalline
        Low-melting eutectic compositions of anhydrous salts

Results

The bonding characteristics of several of the moisture-containing materials were determined. Coatings were prepared using Dixon microfine graphite mixed as a thick slurry in: water solutions of aluminum phosphate or ethyl silicate; water suspensions of Nalco or Baymal; and molten barium hydroxide. The latter, when cooled to room temperature, was a solid mass which was used as a crayon in applying a coating by rubbing on the heated steel sample. The other coatings were spread on the freshly sanded steel surface with a spatula and allowed to dry overnight at room temperature. The solid coatings were then tested by scraping to determine adhesion to the metal, coating hardness, and lubricity. The following observations were made:

1. Aluminum phosphate bond reacted with the steel, releasing H_2 bubbles at the interface. Despite this intermittent interruption, the coating adhered tenaciously to the steel substrate. The bulk of the coating was reasonably strong and polished readily.

2. Silica gel formed from hydrated ethyl silicate did not react with the steel substrate, adhered well and formed a relatively easily scratched coating that polished readily.

3. Silica gel formed from colloidal silica (Nalco) did not react with the steel substrate. The drying shrinkage caused the coating to lift bodily from the metal.

4. Barium hydroxide appeared to have little if any reaction with the metal substrate, formed a strongly adherent coating of reasonable hardness that polished readily. This material reacted with air to form the carbonate and was difficult to apply.

5. Colloidal aluminum hydroxide (Baymal) reacted with the steel, producing a rust-colored-material that permeated the coating. The dried film adhered fairly well to the metal and had a peculiar toughness accompanying a slight difficulty in forming a polished surface when rubbed.

From the above rough screening test, the aluminum phosphate-bonded graphite coating was chosen for further investigation. Four 52100 polished steel flats were used as substrates for testing the friction coefficient of a thin film of the coating in air and in vacuum. Three of these flats were pretreated with a concentrated solution of sodium dichromate and nitric acid to passivate the surface and prevent subsequent reaction with the aluminum phosphate bonding solution. Two coating compositions were prepared: one calculated to contain 40 vol. % solid bond to 60 vol % graphite; the other having 20% bond.

Results from tests on these samples can be seen in the Screening Test Results found in Section VIII.

Glass-Bonded MoS_2 Development

A number of low-melting glasses occurring in the systems A-As-I, S-As-Tl, and S-As-I-Tl were prepared. Such glasses are reportedly capable of wetting and adhering strongly to most metals. Furthermore, some of them are said to be resistant to oxidation, hydration, and acid attack at (and above) room temperature. Because of these properties the sulfur-type glasses were considered as possible bonding media for MoS_2 in thin lubricating films on hardened 52100 steel.

In order to maintain the desired hardness of the metal substrate, if AISI 52100 is used (see Table 25, Section VI, for tempering effects), the temperature for film application must not exceed 400°F (205° C). However, the film itself must have good lubricity and appreciable hardness at the anticipated maximum use temperature of 110° F (43° C). These requirements limit the potentially useful bonding glasses to those compositions possessing relatively low viscosity at 200° C and "softening" temperatures above 45° C. The experimentally prepared glasses were screened on this basis.

Several glass compositions having approximately the desired temperature characteristics were found. These were tested with MoS_2 (Molykote microsize) additions as films applied to freshly sanded steel strips. The resulting coatings were scrape tested to determine relative hardness, lubricity, and adherence to the metal substrate at room temperature. Two of the better coating compositions were selected for application as thin films to the vapor blasted surfaces of hardened 52100 steel test flats. These were used in determining the friction coefficient and wear properties of the films in air and in "vacuum" at room temperature.

Experimental Data

For practical reasons, the low-melting glass compositions were prepared from blends of As_2S_3 (technical grade), sulfur (USP), iodine (reagent quality), and thallium (purified). The mixtures were flame-melted in Pyrex test tubes, loosely capped with aluminum foil after most of the air in the tube had been displaced by dry argon. The observed results differed somewhat from those reported in the literature. The greater viscosity at high temperature, difficulty in obtaining homogeneous melts, and a decided tendency toward preferential volatilization of the several components may be attributable to: the use of the arsenic-sulfur compound (rather than elemental arsenic); the presence of impurities; and the necessity of longer heating at higher temperatures to promote reaction.

Use of pulverized As_2S_3, that had been pre-melted to remove absorbed moisture and consolidate the ultra-fine particles, greatly improved the melting behavior of compositions in the S-As-I system; but none of the glasses containing thallium could be produced with satisfactory homogeneity. The low melting phase that formed (possible Tl_2S or Tl_3AsS_3) resisted further reaction with the remaining constituents. Introduction of thallium into iodine-containing glass seemed to cause a copious evolution of iodine vapor and noticeably increased the viscosity of the glass. Becuase of these undesirable characteristics, the experiments with thallium-containing glasses were abandoned and attention was focused on the more promising S-As-I system. Representative glass compositions tested, and a summary of results are listed in Table 7 while details of the procedures used in forming the glasses are given in Table 8. A Ternary Schematic indicating the compositions tested as well as the glass forming composition area can be seen in Figure 1. The effect of preparation procedure on the behavior of the resulting glass is notably demonstrated by the samples N and N', which, presumably, have identical chemical compositions. The results strongly suggest that true homogeneity (completely random atomic distribution) was not obtained, and that an improved technique must be devised if reproducible results are to be achieved.

The use of these S-As-I glasses as bonding media for MoS_2 was briefly explored as shown by the compositions listed in Table 9, along with the results observed when these mixtures were applied as coatings on freshly sanded steel strips. Preparation and applications procedures are described in Table 8. Although these initial screening tests were not conclusive, the results may be considered as generally characteristic of the behavior of coatings of this type. It was found that:

1. Only those mixtures containing a relatively high glass content were sufficiently soft at application temperature to allow spreading as moderately thin films on heated steel. The surfaces of such films were rough, requiring scraping or burnishing when the material solidified at room temperature.

2. Bond strength within such films at room temperature generally exceeded the film-to-metal bond strength, resulting in the coating flaking off when an attempt was made to remove excessive thickness by scraping.

3. Thin films having good adherence to the steel substrate were obtained by rubbing a pellet of the experimental coating mixture on the freshly sanded (or vapor-blasted) steel surface at room temperature and subsequently heating the coated metal to sinter the applied film. This method of application is suitable for use on the hardened steel races of roller bearings.

4. All of the mixtures tested had low scratch hardness, but appeared to burnish and provide durable thin films when rubbed with a smooth, blunt, steel instrument.

5. The observed brittleness of the pure glasses was notably reduced by the introduction of MoS_2 powder. Consequently, the room temperature behavior of the applied coatings seemed more dependent on the MoS_2 content than on the specific composition of the glass employed.

6. The toughness of the pressed pellets formed from the glass-bonded MoS_2 mixtures suggests that such material may serve as friction-reducing inserts in the retainer cages of roller bearings.

The glasses are readily vaporized at elevated temperatures; but a test made on an "R" glass film held for five hours in a mild vacuum (10^{-3} mm. Hg) at room temperature revealed that no measurable weight loss had occurred.

TABLE 7

GLASS COMPOSITIONS INVESTIGATED IN THE SYSTEM S-AS-I

Sample	Atomic Wt. %			Batch Wt. (gm.)			Observed Characteristics
	S	As	I	As_2S_3	S_8	I_2	
F	60	20	20	2.0	2.8	1.2	Good fluidity when hot; tar-like at room temperature
R	40	25	35	2.06*	1.04	1.75	Good fluidity when hot; brittle at room temperature, but grains fuse together under slight pressure. Poor wetting of steel heated in air at 180°C
P	35	30	35	2.57*	0.79	1.75	Fair fluidity when hot; brittle at r.t. Formed adherent thin layer on steel.
N	32.5	32.5	35	2.46	0.64	1.67	Moderate fluidity when hot; brittle at r.t. Formed adherent thin layer on steel. (Prepared by melting As_2S_3 with S_8, pulverizing, adding I_2 and re-melting)
N'	32.5	32.5	35	2.46*	0.64	1.67	Good fluidity when hot; brittle at r.t. but grains tend to pack under pressure. Softly plastic at 130°C, viscous fluid at 140°C. Poor wetting of steel heated in air at 180°C.
G	30	30	40	2.46*	0.64	2.07	Good fluidity when hot; brittle at r.t.; noticeably softened at 30°C, quite runny at 100°C. Poor wetting of steel heated in air at 180°C
C	26	40	34	3.0	---	1.5	Quite viscous when hot; brittle at r.t.; pulverized readily.
J	20	30	50	3.0	---	3.0	Good fluidity when hot; pulverized readily, but powder coalesced on standing at r.t. Thin layer adheres strongly to steel.

Materials Used

As_2S_3, yellow, technical grade (Matheson, Coleman and Bell)*(Strongly sintered in argon atmosphere, then pulverized before use) (orange color)

S_8, "flowers" of sulfur, U.S.P. (Source Unknown)

I_2, resublimed crystals, reagent quality (Fisher Scientific Company)

TABLE 8

A. PROCEDURE USED IN PREPARING GLASSES

1. Materials were weighed into Pyrex test tube, using an analytical balance.

2. Batch was mixed in the tube, using a small spatula, then packed by tapping the tube.

3. Air in the test tube was expelled and replaced with dry argon introduced near the packed material and flowing at a rate of 20 ft^3/hr.

4. After purging the tube for one minute, the tube was capped with aluminum foil.

5. The tube was gently heated in the flame of a Bunsen burner until the contents appeared to be largely liquid.

6. The tube was tilted and rotated to coat the sides of the tube with the molten material. When again held upright, much of the material adhered to the sides of the tube even after gentle heating.

7. Strong heating was required to remove this material (primarily unreacted As_2S_3) by volatilization, leaving a light gray residue.

8. When the bulk of the material had been gathered in the bottom of the tube it was strongly heated until volatilization occurred, which condensed on the cooler walls of the tube.

9. The sides of the tube were again cleaned by strong heating and the glass recollected in the bottom of the tube.

10. The tube and contents were cooled to room temperature. The aluminim cap was removed, the tube again flushed with dry argon, and re-closed with a small cup formed from aluminum foil firmly held in place by a wire loop.

11. The experimental mixture was re-heated to fluidity and the tube tilted so that most of the glass mixture ran into the cup.

12. When cooled to room temperature, the cup containing the glass was removed from the tube and the aluminum foil peeled away.

13. Most of the glasses prepared had sufficiently high viscosity at room temperature that they could be shattered by a sharp blow, and could be more or less successfully pulverized in a mortar.

TABLE 8 (Continued)

B. PROCEDURE USED IN PREPARING COATING MIXTURES

1. MoS_2 (microsize, Molykote Corp.) was introduced into a Pyrex test tube, lightly packed by tapping, until the tube was filled to within an inch of the top. The tube was capped with aluminum foil, and the contents heated in a Bunsen flame. When water vapor condensed on the upper portion of the tube, the aluminum cap was loosened and the water driven out by heating. The process was repeated until no further condensation of moisture occurred. When cool, the tube was tightly closed with a rubber stopper.

2. The desired quantities of pre-dried MoS_2 and experimental glass powder were weighed on an analytical balance.

3. The components were well mixed together by stirring with a small spatula.

4. The mixture was formed into a 1/2" diameter pellet in a steel die using an applied pressure of approximately three tons.

5. The pellet was tightly wrapped in aluminum foil, excluding as much air as possible, and the package either: (a) placed in a Pyrex tube purged with dry argon and capped with aluminum foil; or (b) given no additional protection.

6. The foil-wrapped pellet was heated to melt the glass and promote homogeneous distribution of the glass in the MoS_2. Those pellets in an argon atmosphere were heated over a Bunsen flame until slight evidence of volatilization of the glass was observed. Some pellets were simply heated in air within a small oven held at approximately 180°C until the foil wrapped package could be easily dented when pinched with tongs, indicating that the material inside was relatively soft.

7. When cooled to room temperature, the foil was removed and the sintered pellet was crushed to produce fine granules.

8. The granulated material was reformed under pressure as a hard, well-bonded, pellet (1/2" diam.)

TABLE 8 (Continued)

C. PROCEDURE USED IN APPLYING COATING ON STEEL

1. Preliminary tests were made using freshly sanded steel strips. These were heated in air within a small oven held at temperatures ranging from 150° to 220°C. The coating was applied by rubbing the pellet, formed from the glass bonded MoS_2 mixture, on the hot steel. Usually the coating would be softly plastic at these temperatures and could be further spread (like butter) with a spatula. The resulting coatings were excessively thick and, when cooled to room temperature, were scraped down with a knife blade.

2. Better adhesion of the coating on the steel seemed to be obtained by rubbing the pellet of the coating mixture on the steel at room temperature. The resulting thin film was then sintered on by heating in the oven at temperatures ranging from 150° to 190°C for several minutes (long enough to heat bearing through). When the steel strip was cooled to room temperature, the process was repeated. Heavier coatings could then be built up by light application of the coating mixture on the hot sample.

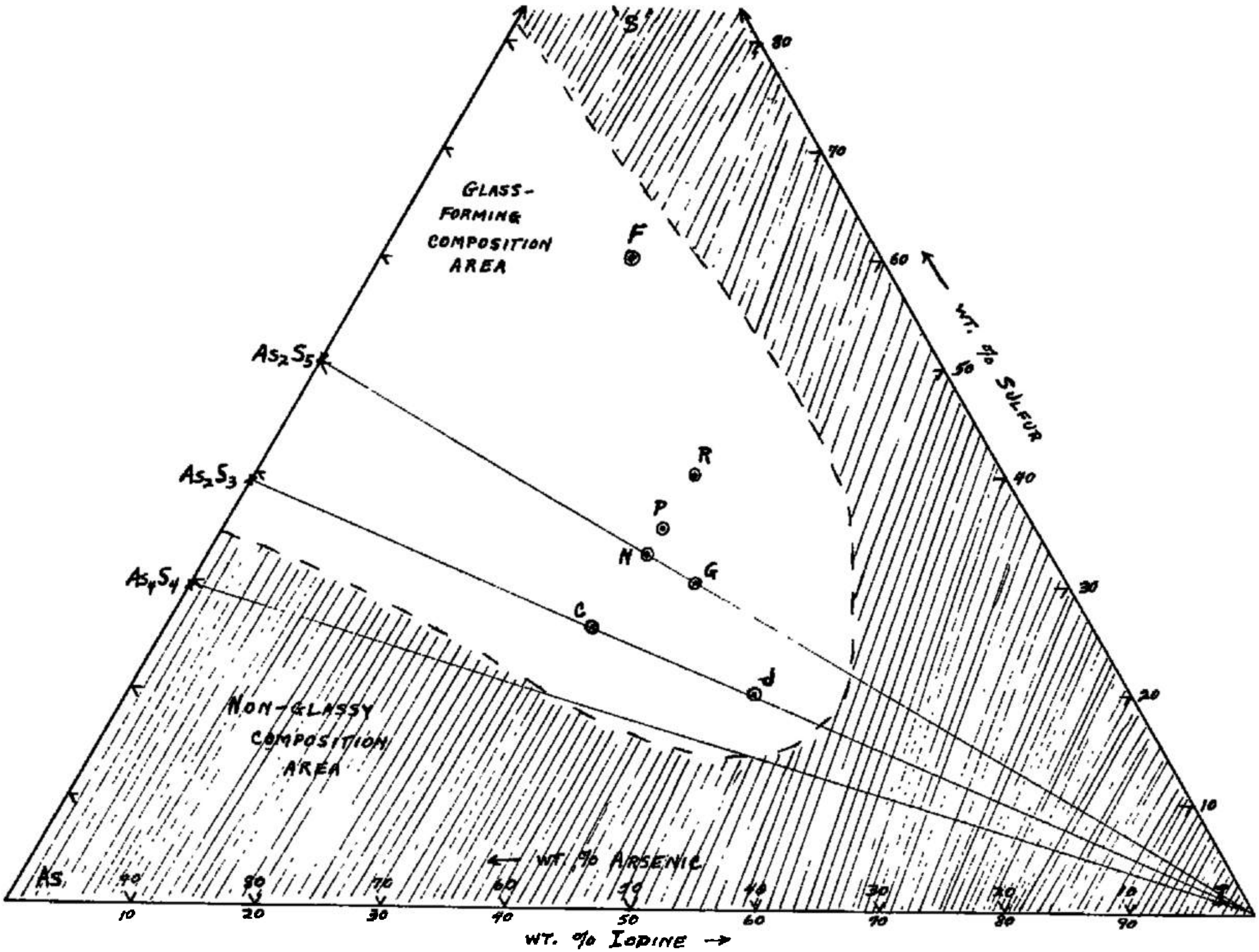

Fig. 1 Ternary Schematic of Glass Binder Compositions Examined

TABLE 9

GLASS-BONDED MoS_2 MIXTURES TESTED AS COATINGS ON STEEL

Sample	Batch Wt. (Gm.) MoS_2*	Glass	Relative Powder Volumes (approx.)	
G	----	---	----	MoS_2 powder mixed into molten glass on heated steel. When cold, excess coating shaves off as powder. Excellent adherence to steel at room temperature.
P	----	---	----	As above
N	----	---	-----	As above
N_1	1.5	(N')1.5	2MoS_2:1 glass	Mixture pre-sintered in argon. Solids content too high for good application on heated steel ($\sim$ 200°C). Poor adherence at room temperature.
N_2	1.0	(N')1.5	> 1MoS_2:1 glass	Prepared as above. Semi-stiff consistency at application temp. (170-180°C). Poor adherence at room temperature. Adherence much improved when film is rubbed on cold steel and subsequently sintered at 170-180°C.
N_3	1.0	(N')2.0	1MoS_2:1 glass	Prepared as above. Soft-plastic consistency at application temp. (170-180°C). Coating can be flaked off when cold. Adheres well when rubbed on cold steel then sintered.
R_1	1.0	(R) 1.0	2MoS_2:1 glass	Mixture pre-sintered at 170°C. Semi-soft plastic at application temp (150-170°C). Coating shaves off as "curls" when cold. Fair adherence to metal substrate at r.t.
R_2	1.0	(R) 1.5	---	Prepared as above. Soft-plastic at 170°C. Very slightly tacky at room temperature.
C_1	1.3	(C) 1.0	> 2MoS_2:1 glass	Mixture pre-sintered in argon. Application temp. well above 200°C promotes volatilization. Coating is stiff-plastic at application temp., adheres well to steel when at r.t.
C_2	1.45	(C) 2.22	1MoS_2:1 glass	Prepared as above. Soft-plastic at application temp. ($\gg$200°C) Coating, brittle at r.t., can be flaked off with knife blade.

TABLE 9 (Continued)

J_1	2.8	(J) 2.0	> $2MoS_2$:1 glass	Prepared as above. Stiff at application temp. due to high solids content, but thin film adheres when pellet of mixture is rubbed against hot steel. When cold, this film adheres well and can be burnished.
J_2	1.5	(J) 2.0	$1MoS_2$:1 glass	Prepared as above. Applies readily to moderately heated steel. At r.t. the coating adheres well but can easily be scratched off with knife blade.

*MoS_2 (microsize, Molykote) pre-heated in argon atmosphere to remove adsorbed moisture and volatiles.

Vapor-blasted, hardened, 52100 steel flats provided with "rubbed-on" thin films of compositions N_3 and R_1 were screened at room temperature in air and in "vacuum" (dry argon) under heavily loaded slow speed sliding conditions. The results of these screening tests can be seen in the Screening Test Results reported in Section VIII. The results of full scale bearing tests using the glass-MoS_2 lubricant can be seen in Section IX and Section X.

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C. ORGANIC LUBRICANT STUDY

A monomolecular layer of an organic material deposited on a metal bearing surface offers attractive possibilities as a solid lubricant. Several mechanisms appear to be available by which the organic can be anchored to the bearing surface. These are: formation of coordination complexes, such as chelates; salt formation, as for example reaction of an organic acid with the metal surface; and adsorption.

To be practical, these organics must have certain characteristics.

a. The film must reduce frictional forces
b. The film must be either very tough or easily replaceable
c. Methods of carrying out required chemical reactions must be adaptable to the treatment of metallic surfaces
d. The coating must be stable in the service environment (hydrolytic stability, etc.)
e. By-products of any required chemical reaction must not be harmful to the bearing.

1. The earliest mode of chemical bonding worthy of consideration in this connection is the reaction between a metal surface and an organic acid. Tingle(1) investigated the lubrication of iron by stearic acid and demonstrated that iron stearate formed at the surface of the boundary layer is the useful lubricant. This reaction is effected by the environment in which it takes place. There is some evidence to indicate that the reaction occurring between organic acids and various metals are different in space than in atmospheric environment.(2)

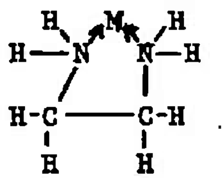
2. Coordination Compounds - Many metals are capable of reacting with organic compounds to form organo-metallic coordination compounds. This discussion is limited to a consideration of metals found in bearings presently being considered for use in space chambers; specifically iron, nickel, and chromium.

   Coordination compounds are generally prepared by fairly complex chemical reactions. Work to date has not been directed toward the preparation of surface films, therefore considerable effort is required to modify some of them for metallic surface treatments. Three reactions(3) utilizing iron and nickel are:

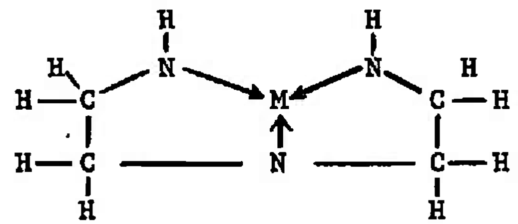
   a. The reaction of C_5H_5MgBr in benzene under an inert atmosphere, usually nitrogen, with the metal.
   b. The reaction of C_5H_5Na in tetrahydrofuran with the metal halide(4).
   c. The reaction of C_5H_5K in liquid ammonia with the metal thiocyanate(5).

   The procedures used to carry out these reactions would have to be modified before surface treatment would be practical.

   One class of coordination compounds, chelates, have been suggested as solid lubricants. Chelates are formed from the union of metal atoms with organic or inorganic molecules or ions and are characterized by having more than one point of attachment to the metal atom. The organic portion is known as a ligand. The following are two examples of the sort of structures possible.



Bidentate Ligand



Tridentate Ligand

Phthalocyanine is a tetradentate plane chelate agent capable of reacting with iron and nickel. It has been theorized by Glasser, et al, [6] that phthalocyanine should be capable of reacting with a metal surface to form a thin adherent film of the corresponding metal phthalocyanine and, since the distance between planes of the phthalocyanine is almost the same as the distance between graphite planes, it was predicted that good lubricating properties would be obtained. These workers found that although phthalocyanine was a good lubricant, copper phthalocyanine was not.

Considering the lubricity of phthalocyanine, the following hypotheses can be presented:

1. It has been suggested that phthalocyanine is a good lubricant because it has a structure similar to graphite, and depends upon the lack of attraction between planes for its lubricity. However, it was shown that the copper phthalocyanine was not a good lubricant. If the mechanism of lubricity were the same as graphite, the presence of the metal atom in the center of the plane should have made no difference, since there is little reason to expect that the metal atom would increase the attraction or bonding between planes.

2. Another mechanism which has been suggested is based on the ability of phthalocyanine to form a chelate with metals. It has been visualized that formation of a chelate of phthalocyanine at a metal surface should result in a low friction surface. While this represents an attractive possibility, it is unlikely that this type of chelation can be achieved with phthalocyanine.

(a) Chelates are normally made from salts of the metal rather than the free metal. For instance, if ferrous chloride is the salt involved, a simple reaction can be shown in which HCl is split out and a coordinate bond is established between the nitrogen of the phthalocyanine and the metal. To start with a free metal and attempt to react it with phthalocyanine requires a two-step process: the first is removal of the electrons from the metal; and the second is reaction of the metal ion with phthalocyanine.

(b) A further complication arises with planar compounds of the phthalocyanine type since the metal atom must lie in the same plane as the organic portion of the chelate. This requires the displacement of the metal atom from the bearing surface. If this occurs, the chelate is obviously not bonded to the bearing.

3. Another mechanism which might account for the observed lubricity is that cf chemisorption. One would expect that metal-free phthalocyanine would adsorb on the metal surface by virtue of donation of the lone pair electrons of the nitrogen atoms. In doing this, phthalocyanine would lie flat on the metal surface and the displacement of a metal atom from the bearing would not be required.

Copper phthalocyanine has substantially fewer of these electrons (by virtue of internal coordination to the copper) and should have a greatly reduced ability to be adsorbed on the surface of the metal, in line with the experimental data.

Another interesting observation was made[6] when it was found that at elevated temperatures the phthalocyanine evaporated from the surface of the bearing. This is precisely the phenomenon one observes in adsorbed systems. From the efficiency of the phthalocyanine as a lubricant, it appears that the phthalocyanine was tightly bound to the surface of the bearing at ordinary temperatures. The range of bond energies obtained by chemisorption (20-100 kilocalories per mole) are adequate to explain the observation.

The arguments presented above should not be considered a proof of the mechanism of the lubricity demonstrated by phthalocyanine, but merely conjecture drawn from the available data. If phthalocyanine is effective by virtue of low intermolecular attraction, one would synthesize other planer materials having even less intermolecular attraction. (Replace the bridge nitrogens with carbonations.) If phthalocyanine is effective by virtue of its ability to form chelates, bidentate and tridentate chelating agents should be investigated. If phthalocyanine is effective by virtue of its ability to chemisorb on a metallic surface, molecular modification should be made which would enhance the bonding. (Additional polar groups should be effective.)

<u>Adsorption</u>

From the available data it would appear the chemisorption is the most likely mechanism. If this is true, certain advantages can be realized.

1. Adsorption takes place instantaneously whereas chemical bonds are slow to form and difficult to establish. Adsorbed organics can be easily replenished as required to replace worn-away material.

2. The energy required for the adsorption reaction is the energy of wetting. This can be supplied as work by running the bearing.

3. Adsorption reactions yield no side-products. Some of the side-products produced by chemical reaction can be abrasive or corrosive.

4. The chemisorption reaction seems adequately versatile. Many combinations of metals and organics are subject to this phenomenon. For instance, steel has been coated with sulfur-containing organics (mercaptans) to provide a surface film[7].

Types of Adsorption

It should be mentioned that there are two main categories of adsorption: van der Waals adsorption and chemisorption.

Van der Waals adsorption is characterized by relatively small energies, in the range of 5 Kcal per mole, and is easily reversible.

Chemisorption involves energies in the range of 20-100 Kcal per mole, and so may be nearly as stable as stoichiometric compounds. This reaction is also reversible and the concentration of adsorbed materials is reduced at higher temperatures. The rate concentration reduction with temperature depends upon the nature of both the metallic surface and the organic. One of the disadvantages of the chemisorption approach is the evaporation of the organic at elevated temperatures. A possibility to negate this effect is the use of polymeric substances rather than the more volatile monomeric organic compounds. Possible candidates for space chamber lubrication applications are the thermally stable polybenzimidazoles.

Gold and other noble metal bearing surfaces are subject to adsorption reactions and can be protected in this manner.

Summary

1. An organic film a few molecules thick bonded to a metal bearing surface is an attractive solid lubricant.

2. Two modes of bonding are possible:

   2.1 Stoichiometric chemical bonding, of which there are two types:

      2.1.1 Salt of an organic acid
      2.1.2 Coordination compounds of which chelates are a special case

   2.2 Adsorption, of which there are two types:

      2.2.1 Van der Waals adsorption involves low energy levels in the range of 5 kilocalories per mole.
      2.2.2 Chemisorption, which involves forces in the range of stoichiometric chemical bonding 20 to 100 kilocalories per mole.

3. Stoichiometric chemical bonding has the advantage of providing films with generally lower vapor pressures than obtainable from adsorbed films.

4. Adsorption has the advantages of instant reaction, easy replacement, no by-products, and versatility with respect to the metal surfaces and the organic components.

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D. PLASTICS

It is realized that the fusing temperature (700° F) for applying films of such organic materials as PTFE on 5200 bearing steel is a limiting metallurgical factor. However, it was felt that because of the outstanding bearing properties of plastics, a brief study should be undertaken to ascertain how well plastics meet simulator environmental requirements and at the same time determine if means could be developed to circumvent the fusing temperature problem.

The following list compares some of the salient conditions to be met in this application with corresponding plastic properties:

Conditions	Properties of Plastics
1. Efficient lubricant	Low coefficient of friction and resistance to wear
2. Ultra-high vacuum	Negligible outgassing
3. Normal atmospheric conditions	Oxidation resistant
4. Radiation environment	Small change in properties at incident radiation dose rates
5. Low temperature environment	Will not shrink or crack at low temperatures
6. High humidity environment	Hydrolytically stable and physical properties unaffected by moisture
7. Used as thin films	Easily applied to complex surfaces without the need for special equipment
8. Used in large quantities	Prepared from commercially available materials

The properties of organic materials depend on their chemical nature and molecular geometry. For the conditions encountered in the application, plastics are the only organic materials to be considered. The following discussion is based on a description of the molecular structure of polymeric materials (plastics) and the effect of this structure on some of the properties.

1. Description of Polymers

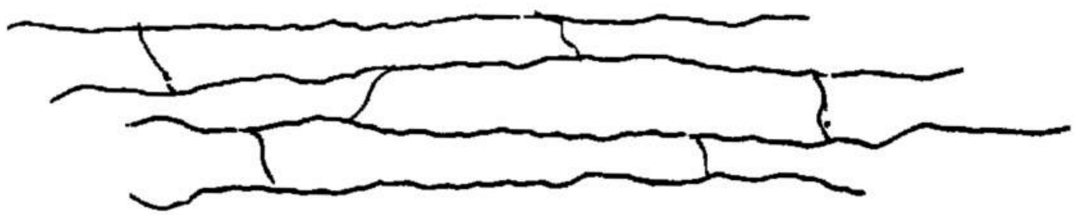
Basically, polymers are built by linking monomers (small organic molecules) together to form chains, often thousands of units long. At the present time, there are 40 or 50 readily available monomers capable of producing an almost limitless number of combinations. These chains can be considered to fall into two classes:

a. Chains unattached to each other, held together by intermolecular attraction and chain entanglements.

b. Chains chemically attached to each other at several points.

The long, unattached chain structures characteristically flow when heated and are described as thermoplastic. Examples of this sort of material are: polyethylene, polypropylene, Teflon and polyvinylchloride.

The attached chains are formed by "crosslinking" special long chain polymers which contain chemical groups specifically incorporated for the purpose. The crosslinked or cured plastic is a three-dimensional network and is described as thermosetting. Examples are: epoxy resins, polyesters, silicone resins, and most kinds of rubber.



Idealized Random Crosslinked Structure

The descriptions given above define the generic differences between major polymer classes but fail to provide complete pictures. The subtleties of polymer structures and compositions can, and do, significantly affect several functionally important variables. Several of these will be touched on in the ensuing discussions.

2. Effect of Molecular Weight Distribution

In the preparation of polymers, one obtains a statistical distribution of chain lengths, referred to as the molecular weight distribution of the polymer. A characteristic curve is shown in Figure 2.

The mechanical behavior of plastics is greatly affected by the molecular weight distribution. One of the properties significant to space chamber applications greatly influenced by molecular weight distribution is the vapor pressure. Each molecular weight fraction has its own vapor pressure; therefore, it is incorrect to speak of the vapor pressure of a plastic. The relationship of vapor pressure to molecular weight shown in Figure 3 is based on extremely low molecular weight materials related to polyethylene.

It can be seen from Figure 3 that the vapor pressure of a polymer having a molecular weight of approximately 350 is .1 mm of mercury. Increasing the molecular weight to about 500 reduces the vapor pressure five orders of magnitude. To go one step further, the polyethylene, which one normally encounters, has an average molecular weight in the range of 50,000. It has often been said that plastic materials cannot be used in high vacuum systems due to the vapor pressure; however, it is obvious that fractionation of the polymer to remove the low molecular weight constituents may eliminate this problem. Methods of accomplishing this are known, and need only be modified for specific plastics.

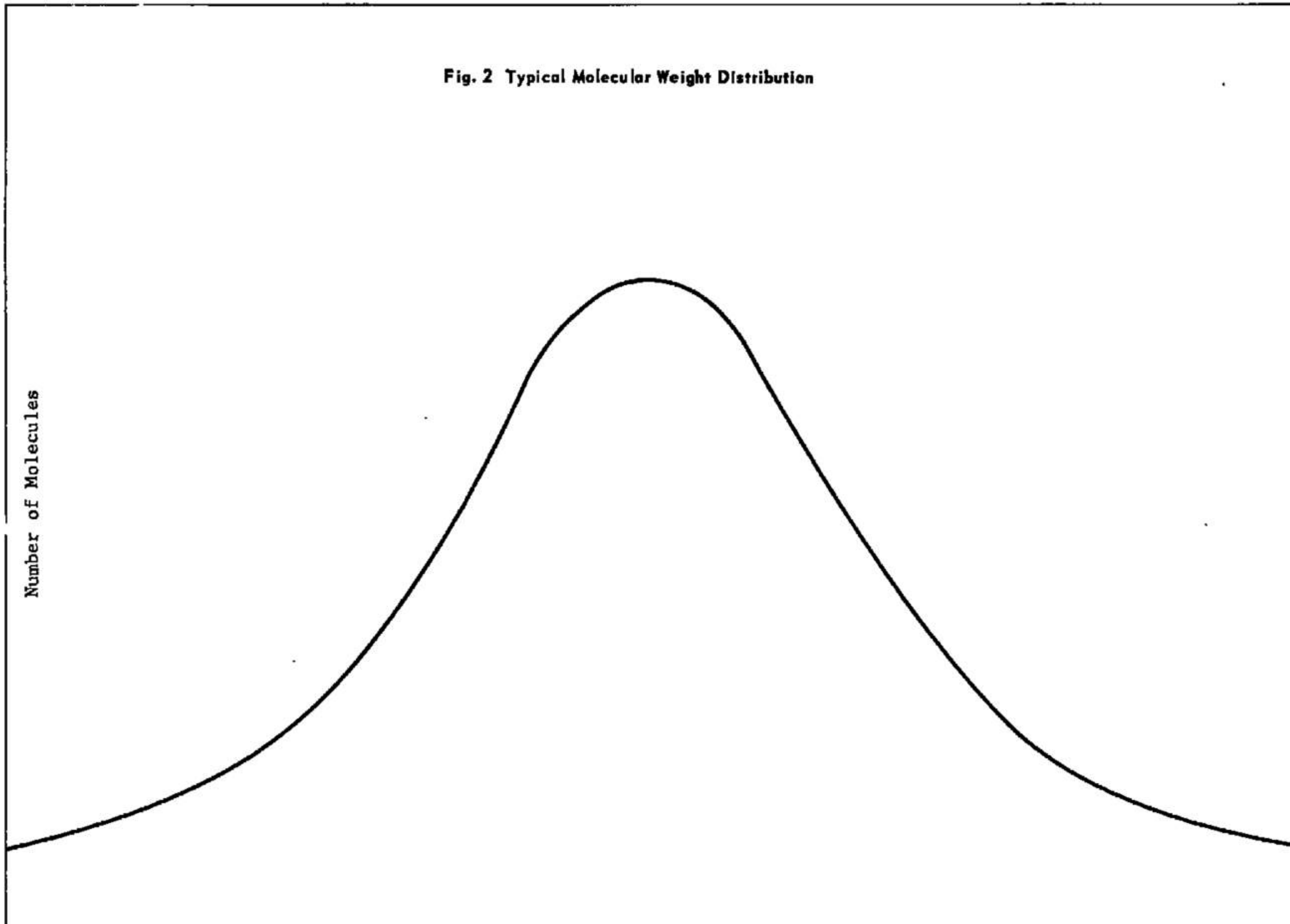


Fig. 2 Typical Molecular Weight Distribution

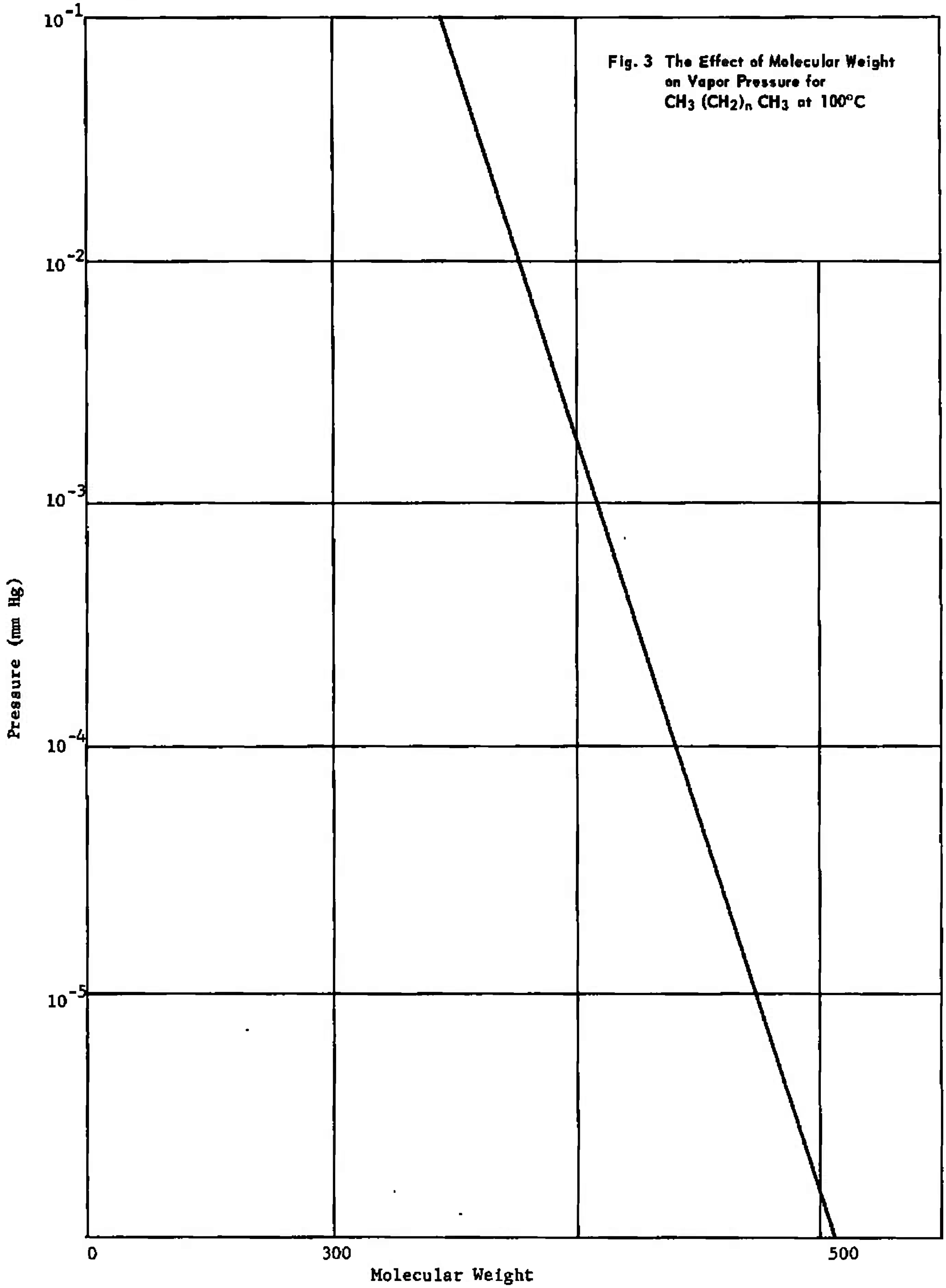


Fig. 3 The Effect of Molecular Weight on Vapor Pressure for $CH_3(CH_2)_nCH_3$ at 100°C

Future work should include a study of the removal of low molecular weight fractions from useful organic solid lubricants. It is interesting to note that certain lubricants, such as silicone SF 50, can be fractionated to yield a fluid lubricant having a vapor pressure below the measurable level.

The vapor pressure of an organic compound is also a function of the temperature. Figure 4 is a typical curve relating vapor pressure to temperature for a pure organic compound.

Another interesting feature of the molecular weight is its effect on the viscosity of a thermoplastic material. For instance, two silicone fluids having essentially the same structure, except for molecular weight, (chain length) can vary in viscosity from the fluidity of water to a material still technically a fluid but with a viscosity so high that an air bubble will rise through it at the rate of only 1 cm per year. One of the most profound influences on the mechanical behavior of a given plastic is the molecular weight.

3. Effects of Mechanical Action on the Molecular Weight of Polymers

It is a consequence of the unique long chain structure of polymer molecules that purely mechanical forces are capable of fracturing them. This action can be brought about by shaking,[1] beating,[2] high-speed stirring,[3] or turbulent flow.[4]

The rate of degradation is effected by temperature and, in some instances, oxygen increases the rate.[5] To determine the significance of the effect, the study of the phenomenon should be directed toward the specific polymer end application.

4. Radiation Effects on Molecular Weight

Some space chamber applications are concerned with radiation conditions. Radiation effects on plastic materials result from two reactions. These take place simultaneously; however, one or the other is usually favored depending upon the plastic material and other circumstances.

The first reaction is one which crosslinks the polymer chains, and is one method of increasing the molecular weight of a polymer. Polyethylene can be crosslinked in this fashion to leading a new material having a higher heat distortion temperature than the starting material.

On the other hand, polypropylene favors the alternate reaction, which is a random chain scission and results in decreasing molecular weight. A useful example of the first reaction is described by Hartzband, Tarmy and Long,[6] in which hexadecamine is subjected to a radiation dose of 3×10^8r to yield a high viscosity lubricating oil.

5. Plastics as Lubricants and Bearings

A great deal of data has been collected with regard to the use of plastics as solid lubricants and bearing materials. Unfortunately, most of this work has been directed toward the development of materials for specific applications

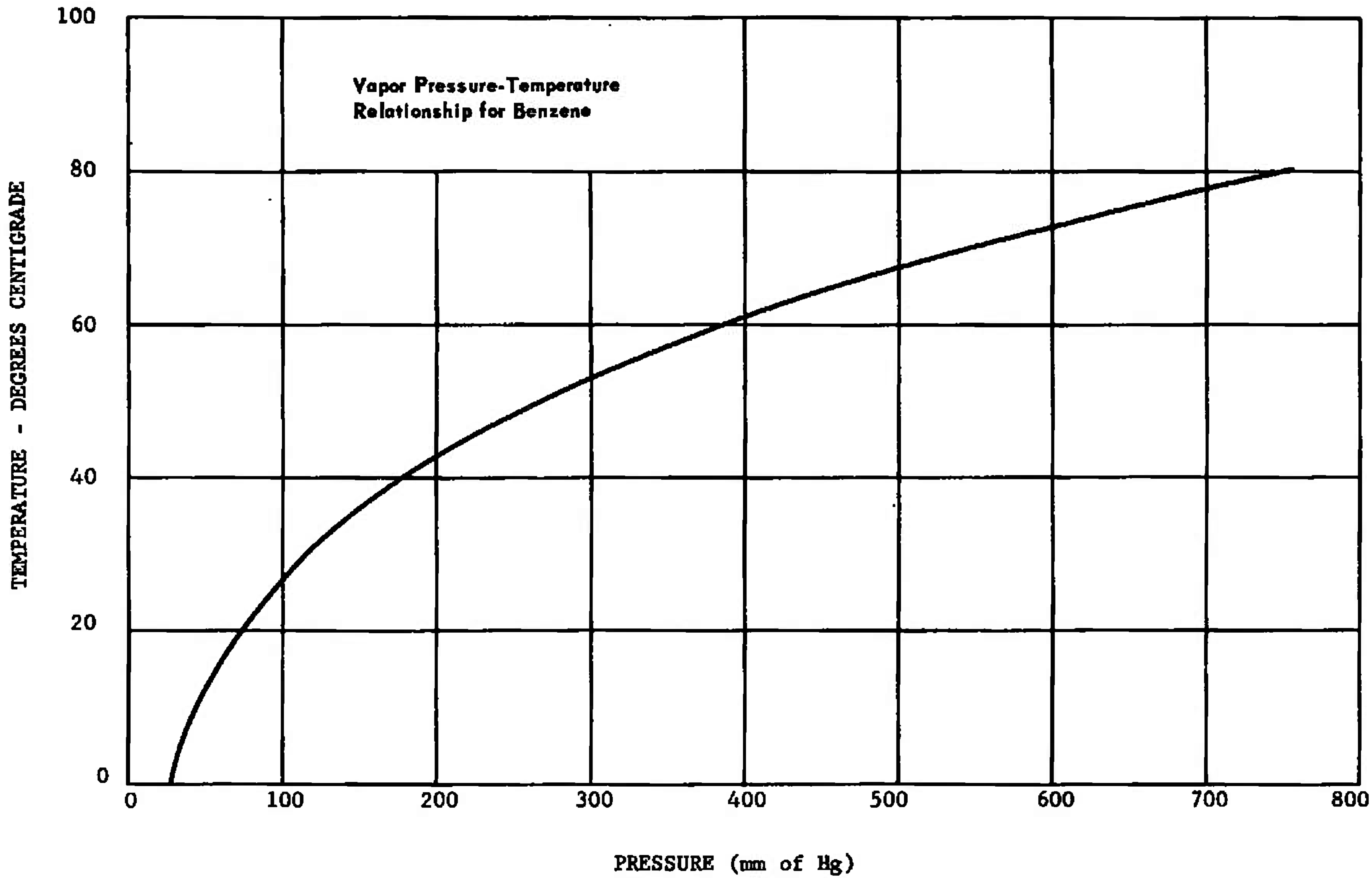

Vapor Pressure-Temperature
Relationship for Benzene
TEMPERATURE - DEGREES CENTIGRADE
100
80
60
40
20
0
0
100
200
300
400
500
600
700
800
PRESSURE (mm of Hg)

Figure 4

and has not yielded much fundamental information correlating the physical chemistry of polymers with their efficiencies as solid lubricants and bearings. At the present state of the art it is difficult to predict the usefulness of these materials. Plastics can play two distinctly different roles in lubricant and bearing systems: the organic can serve as the exclusive lubricant, or it can serve as a binder holding some other material, such as molybdenum sulfide, which provides the lubricating properties. The simpler case is the first -- the organic serving as the lubricant. Plastic lubricants and bearings have been used in three forms:

a. Plastic has been used by itself as the exclusive lubricant
b. Plastic has been reinforced with fabrics
c. Metal bearings have been impregnated with plastics

Polytetrafluoroethylene (Teflon) and polyamides (nylon) have been used in all three of the above systems. Phenolic resins have served as self-lubricating bearings, but only in conjunction with a fabric or paper for reinforcement. It is interesting to note that nylon is often used as the reinforcing fabric for phenolic laminates for these applications.

6. The Mechanistic Basis of Low Friction

Inorganic lubricants such as molybdenum sulfide and graphite are good lubricants. The crystals are flat plates attached by extremely weak bonds. The forces are so low, in fact, that the plates can easily slide across each other. The mechanism of low friction in plastics is much different and much more complex. Different plastics can be used as bearings and lubricants by virtue of different mechanisms. Allen [7] suggests that the success of nylon as a bearing surface is due mainly to its resistance to wear rather than to a low coefficient of friction. Teflon, on the other hand, functions more in the manner of a true lubricant. King and Tabor [8] have shown that Teflon does not have a surface film in the usual sense, but the lubricity of Teflon results from low intermolecular cohesive forces between adjacent chains. This may be surprising since strong dipoles, such as the carbon fluorine dipole, tend to promote strong intermolecular attraction. Apparently, the comparatively large fluorine atoms screen the dipole sufficiently to negate the usual effect. At a frictional surface the sheering forces are, of course, the greatest and can easily overcome the weak intermolecular forces. On the other hand, within the bulk of the Teflon, sheer forces are much less and the major contributors to sheer strength, specifically chain entanglements and crystalline forces resulting from highly oriented chains, are effective.

It is well known that one of the factors in wear is the temperature. It has been shown by Zisman, et. al., [9] that a thin film of Teflon upon metal provides a useful system as the result of the metal providing a heat sink.

7. Influence of Bulk Polymer Properties on Friction

The work of Tabor has provided evidence that the elastic hysteresis losses in rubber are important for determining the effect of coefficient of rolling friction[10-14]. This phenomenon was investigated in more detail by Bueche and Flom [15] who found that the change of friction with speed for steel on Plexiglas lubricated with sodium stearate exhibited a behavior similar to mechanical loss vs. frequency. These investigators further noted that the frictional force required to make steel slide on branched polyethylene was greater than on unbranched polyethylene. Kline, Sauer and Woodward [16] measured the effect of branching on the dynamic mechanical properties of polyethylene and found that at room temperature, and above, a decrease in branching was accompanied by a decrease in damping as would be expected from the work by Bueche and Flom. Bueche and Flom further found that the relationship between rolling friction and speed for a hard sphere of constant diameter on softer material for a given retardation time closely resembles the relationship between mechanical loss and frequency.

It would appear, therefore, that anything that would affect the loss tangent of the material will affect its frictional characteristics. Woodward and Sauer [17] note that in many materials, including Teflon, that the loss tangent depends on temperature. Loss peaks occur at various temperatures and can be attributed to changes in the structure of the polymer, such as crystallization.

8. Filled Plastics

Many applications for which plastics are eminently suited have one or more requirements beyond the normal scope of the plastic. To satisfy these unusual requirements, fillers may be added. These can be minerals, such as clay; fabrics, such as cotton or glass; or other plastics, such as finely divided rubber used to impart crack resistance to polystyrene. Plastics modified with fillers have been found useful in the field of lubrication and bearings. Table 10 compares some properties of Teflon filled with various materials [18]. These filled Teflons illustrate some of the important parts of plastics which are of interest to bearing and lubricant applications and which can be modified by the addition of a filler. (Tables 11 and 12). Table 13 shows the coefficient of friction as a function of speed and also some relative wear data for bronze-filled Teflons.

9. General Fabrication Methods

An important consideration in the choice of plastic materials is that of the limitation introduced by the method of fabrication. The following are brief descriptions of two important methods of fabricating plastic bearing surfaces:

(a) Bearings molded to shape

Plastic parts can be accurately fabricated to very close tolerances by molding under pressure. The techniques are versatile and well established and are suitable for use with either filled or unfilled plastics. A major disadvantage lies in the difficulty of design and the expense of fabrication of the necessary molds. A complete description of the techniques of compression molding can be found in any plastics handbook [19].

(b) Plastic-coated metal bearings

It has been previously mentioned that a thin coat of plastic on a metal bearing provides an excellent low friction surface. This

TABLE 10

THE COMPRESSIVE STRENGTH OF FILLED P.T.F.E. MATERIALS

Material	Filler by volume, per cent	Modulus x 10^2, lb/in^2	Compressive strength, lb/in^2
P.T.F.E. + bronze	30	445	2405
P.T.F.E. + bronze	40	695	2600
P.T.F.E. + bronze	50	941	3775
P.T.F.E. + Kieselguhr	30	718	3320
P.T.F.E. + Kieselguhr	40	836	4100
P.T.F.E. + MoS_2	30	586	2955
Bronze fibre	30	845	4675
Nylon	--	1355	6410
Pure P.T.F.E.	--	810	1775

TABLE 11

THE THERMAL EXPANSION OF FILLED P.T.F.E. MATERIALS

Filler	Filler by volume per cent	Linear Expansion coefficient at 100° C, 10^{-5} in/in/° C
Pure P.T.F.E.	--	13.7
Bronze	30	9.5
Bronze	40	9.1
Bronze	50	7.4
Kieselguhr	30	8.9
Kieselguhr	40	7.2
MoS_2	30	10.2
MoS_2	40	8.1

TABLE 12

THE EFFECT OF BRONZE FILLER ON P.T.F.E. MATERIALS

Material	Thermal conductivity cal cm^{-1} sec^{-1} $°C^{-1}$ x 10^{-3}
P.T.F.E.	0.57
30 per cent Kieselguhr	0.98
30 per cent bronze	1.08
50 per cent bronze	1.70
30 per cent MoS_2	1.63

TABLE 13

THE THERMAL CONDUCTIVITY OF FILLED P,T,F,E, MATERIALS

Material Vol. % Bronze	Coefficient of Friction(1) f Initial Value	4 Hrs.	20 Hrs.	Relative Wear Vol.(2) in^3 x 10^{-10}
30	0.07	0.17	0.20	0.8
40	0.08	0.17		1.0
50	0.09	0.17	0.19	1.7

(1) Load: 4.15 lbs.
Vel: 2.62 ft/min

(2) Load: 16 lbs.
Vel: 113 ft/min
Period: 4 hrs.

construction tends to reduce the overheating problem which one faces with all-plastic bearings because the metal bearing performs the function of a heat sink. Teflon-coated bearings can be fabricated in the following manner, for example:

(1) Clean the metal bearing surface
(2) Coat the bearing with Teflon emulsion
(3) Air dry
(4) Sinter the plastic above its softening temperature.

Certain problems are encountered with the use of this technique. Some of them are as follows:

(1) It is difficult to obtain a coating of uniform thickness, especially on complex shapes
(2) Sintering temperature will adversely affect some steels
(3) Coatings of less than a mil are difficult to obtain and often contain pin holes.

The advantages of this approach are:

(1) Filled plastics can be used
(2) Bearings prepared in this fashion will operate at higher speeds than all-plastic bearings.

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SECTION IV – RADIATION EFFECTS ON MATERIALS AND LUBRICANTS

INTRODUCTION

It is highly probable that there will be times when a nuclear auxiliary power unit will be operating within a simulator chamber as part of the test vehicle. The general nature of the nuclear reactor is described in item 3 of the footnotes found in Table 1, Section I, "Typical Space Simulator Applications."

Because of the possibility of radiation damage to materials and lubricants, a brief analysis was made to enable the chamber user to have some idea as to radiation types and levels that may be expected at specific distances from the radiation source. Having this information in addition to anticipated periods of exposure, as well as available information from the literature on the effects of type and dosage of radiation on the material in question, a conclusion can be drawn as to the correctness in choice of lubricants and materials.

RADIATION EFFECTS

1. Factors in Radiation Damage

Radiation damage may be defined as the changes in the properties of a material which has been subjected to nuclear radiation. Property changes in materials do not occur at the same time, direction or at the same rate. Although not all such changes are detrimental for all applications in which the materials might be used, they too are considered to be damaged in the sense that the properties differ from those of the unirradiated materials.

Radiation damage in organic materials result from the formation of foreign compounds in the material. As nuclear particles traverse a material, they transfer energy to the electrons and nuclei of individual atoms in quantities sufficient to break the bonds or linkages which bind the atoms into molecular groups. Following the passage of the radiation particle, the fragments of the disrupted molecules react chemically to form compounds which differ from those originally present. It follows that the concentration of these impurity compounds increase with increasing amount of radiation and results in correspondingly greater change in the material's properties.

Structure metals and alloys undergo changes in their mechanical properties during irradiation in a reactor environment. The changes produced by irradiation have been ascribed to an atomic displacement mechanism in which the bombarding neutrons collide with an atom within the metal lattice. The knocked-on atom comes to rest within an interstitial position after expending its energy. This event is repeated numerous times during irradiation. The degree of changes in the properties has been shown to be a function primarily of the total number of events occurring, the longer the exposure, the greater the change in properties.

2. Kind of Radiation

The kind of radiation to which material is subjected affects the extent of damage by the manner or process by which a particle transfers its energy to a material, the important factor being that not all classes of material are damaged by all of the energy transfer process.

To cover a lot of ground, the various particles transfer energy to materials by interaction with either electrons or nuclei of atoms. Those particles which transfer energy to electrons produce heat but do not produce extensive damage in some materials, for example, metals and alloys. The particle which interacts with the nuclei of atoms produces permanent damage in all materials. Organic materials, in contrast to metal and alloys, are damaged by any particle or energy transfer process, and it is this factor which places the organic materials at the bottom end of the scale of radiation stability.

The types of radiation particles and their interaction processes are briefly summarized in Tables 14 and 15. Of these, neutrons and gamma photons, because they are the only kinds of radiation that can penetrate more than a few centimeters in solid matter, comprise the radiation field in and about nuclear reactors. These two types, however, produce all of the other types of particles in the materials which they traverse; it is through the action of these secondary, short-range particles that damage is produced.

3. Rate of Irradiation

It is generally accepted that dosage rate is not an important factor in radiation damage when other environmental factors such as time, temperature, or atmosphere do not contribute significantly to deterioration. Dosage rate must be considered when radiation damage is accompanied by significant deterioration of the material from other environmental factors. In the irradiation of an organic liquid, for example, the possibilities for oxidation are enhanced due to the presence of molecular fragments which can readily combine with oxygen; damage will therefore be some function of dosage rate rather than dosage alone.

4. Composition of the Materials

Because the probability of interaction of the radiation particles varies for different elements, the composition of a material also affects the extent of damage that occurs from a given amount of radiation dosage.

The absorption of thermal neutrons in some elements can lead to the deposition of very large amounts of energy in some materials. Nearly all elements release an energetic gamma photon upon absorbing a neutron, and while the total energy released in this manner can be very large, most of the energy released escapes from small volumes of material because of the great penetrating ability of gammas. Neutron absorption reactions of importance in some types of organic materials occur in boron, lithium, nitrogen. In these reactions an alpha particle or a proton is released; and since these particles can travel only a few thousandths of an inch, their entire energy is dissipated in the

TABLE 14

TYPES OF RADIATION PARTICLES IN NUCLEAR REACTORS

Primary Particles	Interaction Process	Secondary Particles	Energy Transfer Process
Fast Neutrons	Collision with atomic nuclei (elastic collision)	Recoil atoms Degraded neutron	Ionization and excitation by interaction with orbital electrons of atoms Transfer of kinetic energy to atoms by repulsive interaction of electric fields
	Absorption in atomic nuclei (inelastic collision)	Recoil atoms or gamma photon	Ionization by gamma processes
Thermal Neutrons	Absorption in atomic nuclei to form radio-active atoms	Gamma photon	Ionization by gamma processes
		Alpha particle Proton	Ionization Excitation Transfer of kinetic energy to atoms by a repulsive interaction of electric fields
		Beta particles	Ionization and excitation by interaction with orbital electrons of atoms
Gamma Photons	Photoelectric absorption	Electrons	
	Compton scattering	Electron Degraded gamma photon	Ionization and excitation by interaction with orbital electrons of atoms
	Pair production	Positron (positive electron) Electron Degraded gamma	

TABLE 15

RADIATION UNITS

Unit	Definition	Remarks
ROENTGEN	Amount of ionizing radiation (gamma and X-rays) which imparts 83 ergs per gram of air.	The roentgen, by definition, specifies both the type of radiation and the material, hence it is an indirect means of describing a radiation field comprised of X-rays and gamma photons. Because of differences of various elements in absorbing energy from X-rays or gamma rays, one roentgen of radiation will deposit from 83 ergs per gram (air, Teflon) to nearly 100 ergs per gram (polyethylene) in different materials.
rep (roentgen equivalent physical)	Amount of radiation of any type which imparts 93 ergs per gram of animal tissue.	The rep, by definition, specifies only the material; hence, it is an indirect means of describing a radiation field comprised of any type of nuclear particle - including neutrons, alpha particles, and protons, as well as X-rays and gamma photons. As with the roentgen, one rep of radiation will deposit different amounts of energy in different materials. For greatest utility, radiation fields described by rep should specify the fraction of total rep due to each type of particle.
rad	The absorption of 100 ergs of energy per gram of material from radiation particles.	The rad specifies only an amount of absorbed energy irrespective of the material; hence, it cannot be used to describe a radiation field.
rem (roentgen equivalent man)	The amount of radiation of any type which has the same biological effectiveness as one roentgen of X-rays or gamma radiation.	The rem involves the use of factors for the relative effectiveness of various particles such as protons, alphas, and neutrons as compared to X-rays or gamma rays in producing biological damage. Although the rem can be used as an indirect means of describing a radiation field, its use in materials - damage work is limited.

TABLE 15 (Continued)

Unit	Definition	Remarks
nv	The nv specifically refers to n neutrons per cubic centimeter moving with a velocity v centimeters per second in a given direction; hence the product nv equals the number of neutrons of velocity v which traverse one square centimeter per second. Where v is not otherwise specified, nv is usually considered to apply to thermal neutrons.	The nv is a direct description of the number or flux of neutrons of near-thermal energies present per second in pile radiation. When nv is used, it is implied that the number per second of fast neutrons and gammas present in the particular reactor referred to also pass through the unit area.
nvt	The total number of thermal neutrons which impinge on an area of one square centimeter oriented in any direction in a nuclear reactor in t seconds.	The nvt is the dosage unit corresponding to the nv flux unit and is equal to the product of nv and exposure time in seconds. It is a direct description of the total number of neutrons of near-thermal energies which impinge on an area of one square centimeter; but, as with nv, it is implied that the associated number of fast neutrons and gammas present in the particular reactor referred to also impinge on the square centimeter. Because the nv and nvt specifically describe the thermal neutrons and only imply that fast neutrons and gammas are present, their use in materials-damage work is limited.

material. Such reactions are of considerable importance inside nuclear reactors because of the high proportion of number of thermal neutrons relative to the number of gammas or fast neutrons. Even in radiation fields in which the thermal neutrons are less numerous than the other primary particles, reaction of this type can impart significant amounts of energy to the material.

5. Radiation Field Effects

Essentially the nuclear reactor in question is assumed to be a 45 cm diameter sphere and the following approximate fluxes are associated with it:

Fast neutron	5×10^{12} n'/cm^2/sec. (10^5 ev)
Epithermal neutron	5×10^{12} n'/cm^2/sec. (0.1 ev - 100 ev)
Thermal neutron	5×10^{12} n'/cm^2/sec. (0.025 - 1.0 ev)
Gamma photons	2×10^{13} mev/cm^2/sec (10^4 - 10^7 ev)
	(av. E = 1 mev/cm^2/ sec)

In this area, to simplify the matter, only the neutrons and the gamma or electromagnetic rays will be discussed. From previous discussion it will be remembered that there are several other constituents of the complex radiation generated and their combined effects will accumulate as exposure is prolonged. Figures 5 and 6 give an approximation of intensities of neutrons and gamma photons anticipated as the distances from the source increase. Arrival at these intensities was calculated by using the distance provided (reference Table 1, of Section I,"Typical Space Simulator Applications, and the inverse square law).

6. Metals

It is well known that structural metals and alloys undergo changes in their mechanical properties during irradiation in a reactor environment. The changes produced by irradiation have been ascribed to an atomic displacement mechanism in which the bombarding neutrons collide with an atom within the metal lattice, knocking it from its normal site, thus creating a vacancy within the lattice. The knocked-on atom comes to rest within an interstitial position after expanding its energy. This event is repeated numerous times during irradiation. The degree of change in the properties has been shown to be a function primarily of the total number of events occurring, i.e., the longer the exposure, the greater change in properties. It has also been noted that the greatest rate of change occurs in the early stages of irradiation. Many significant changes have been noted to have reached a degree of saturation after exposure of about 10^{18} - 10^{19}n cm^{-2}. It should be noted that many of these changes in properties are helpful to the designer in that strength of the materials is increased. The limiting property in many instances is the decrease in the ductility accompanying irradiation. The properties of structural materials that are influenced by radiation are

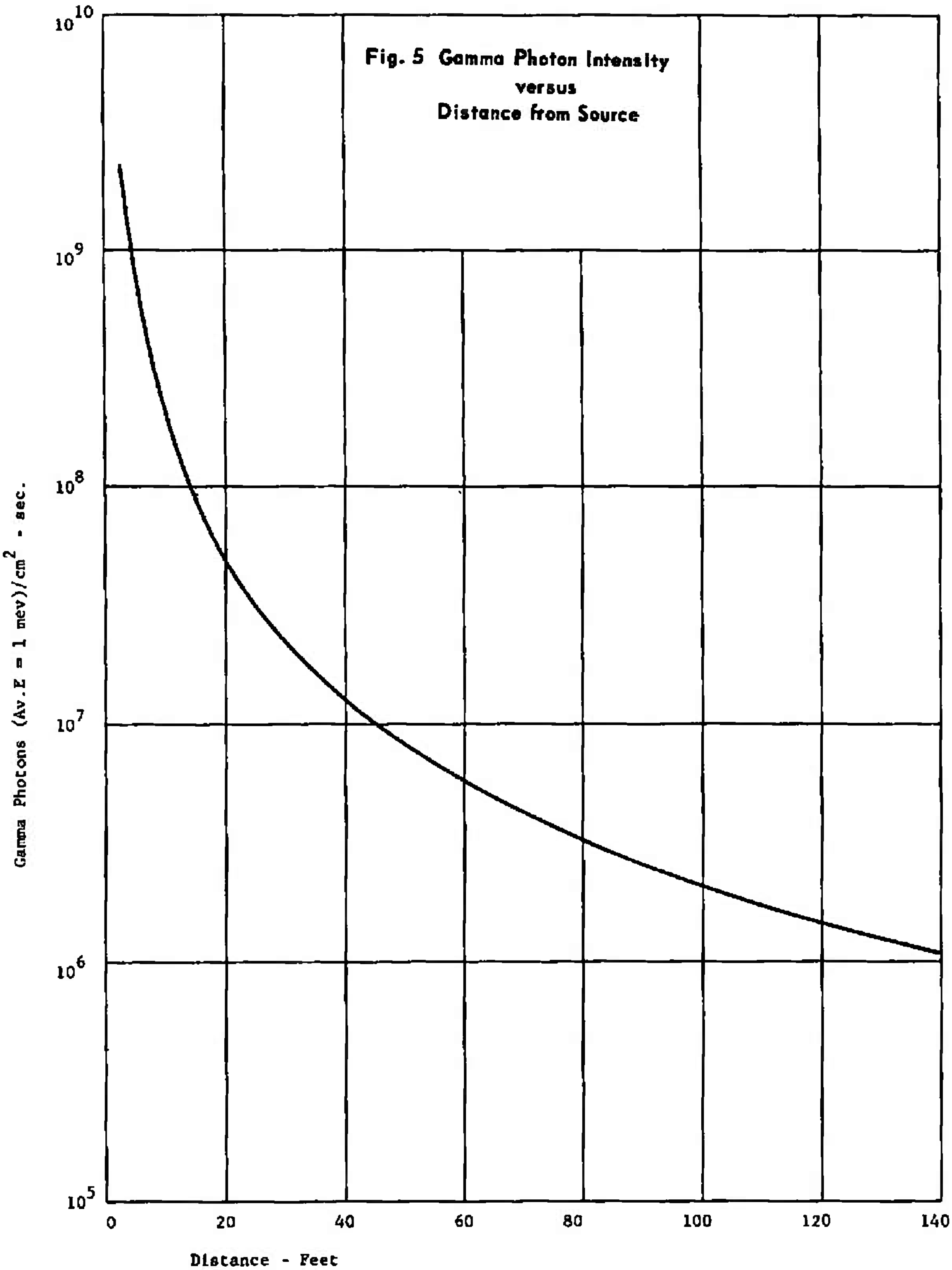


Fig. 5 Gamma Photon Intensity
versus
Distance from Source
10^{10}
10^{9}
10^{8}
10^{7}
10^{6}
10^{5}
Gamma Photons (Av.E = 1 mev)/cm^2 - sec.
0
20
40
60
80
100
120
140
Distance - Feet

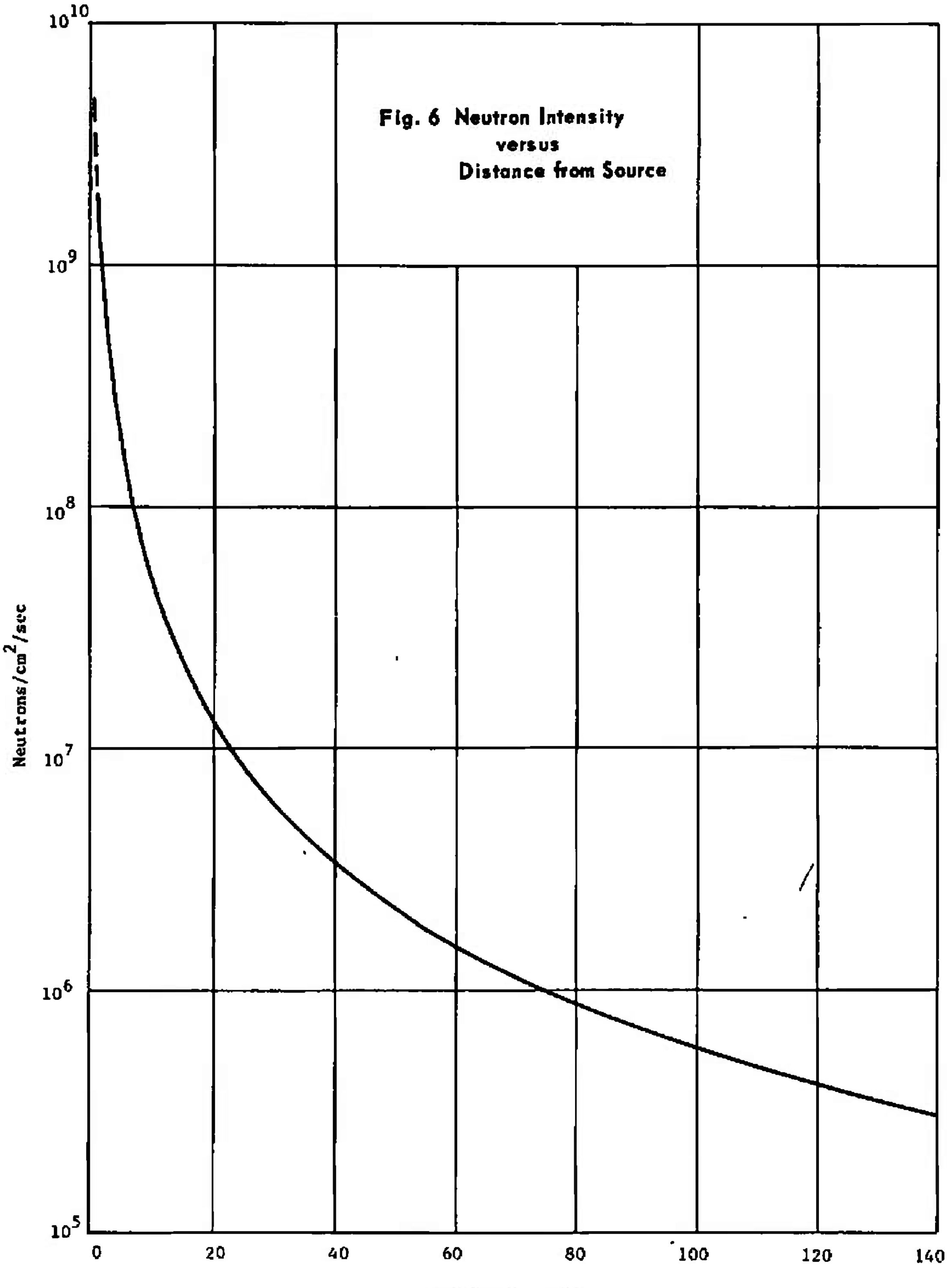


Fig. 6 Neutron Intensity
versus
Distance from Source
10^{10}
10^{9}
10^{8}
10^{7}
10^{6}
10^{5}
Neutrons/cm^2/sec
0
20
40
60
80
100
120
140
Distance - Feet

tensile properties, creep properties, fatigue properties, impact-energy transition temperature, thermal conductivity, and corrosion.

In the area of design data (pressure-vessel steels for reactors which consist primarily of carbon and low-alloy steels), recent studies by the Naval Research Laboratory indicate that high-temperature annealing will remove or lower the brittle-to-ductile transition temperature after exposure to about 10^{20} n cm^{-2}. The removal, though not complete, is effective and represents a recovery of about 90 percent.

7. Lubricants

Organic materials consist primarily of carbon and hydrogen bound together by chemical bonds that are relatively easy to break with the addition of energy. Both gamma rays and neutrons can cause molecular changes which will greatly affect the properties of the material. Some of these materials are shown in Tables 16-19* where they merit-rated in order of decreasing resistance to radiation.

8. Phenolics

Unfilled phenolics stand fairly low in radiation resistance, their tensile and impact strengths decreasing about 50 percent at 10^{10} ergs g^{-1} (C). When irradiated they swell, become very brittle, and tend to crumble. Also, a soluble product is formed which causes the material to disintegrate in water. The addition of fillers, particularly mineral fillers, will increase the stability of phenolics.

9. Silicones

Silicone oils on being irradiated by gamma radiation are converted into solid elastomers. The low molecular weights yield clear, transparent solids which crumble into soft gels on handling. The higher molecular weights yield elastomers having a low tensile strength. It has been shown with silicone polymers as well as with others that the introduction of aromatic groups into the polymer structure greatly increases the stability of these materials to radiation.

10. Inorganic

Solid film lubricants continue to be very promising in an irradiation environment. A solid film consisting of 71 percent molybdenum disulfide, 7 percent graphite, and 22 percent sodium silicate had a bearing performance life of 17 hours at 10^4 rpm and 350 F after a gamma exposure of 10^{11} ergs g^{-1} (C). The wear life of MoS_2, $Na_2B_4O_7$ (6:1), and lubricant A in diaxane (1:3) was tested after gamma exposure up to 10^9 ergs g^{-1} (C) for lubricant A. The wear life of irradiated molybdenum disulfide sample was between 0.646 and 1.175 times the wear life of unirradiated control sample with a 95 percent confidence level. The lubricant A sample had a wear life of between 0.853 and 1.726 times the wear life of unirradiated sample with a 95 percent confidence level.

* Refer to: L. Bupp, L. Burger, R. Harrington, "How Radiation Affects Organic Materials". General Electric Company Information.

TABLE 16

GENERAL EFFECTS OF RADIATION ON ORGANIC COMPOUNDS

Class of Compounds	Products - In Approximate Decreasing Order of Abundance
Saturated hydrocarbons	H_2, CH_4, higher fragments unsaturated hydrocarbons
Unsaturated hydrocarbons	H_2, CH_4, polymers, fragments
Aromatic hydrocarbons	Polymer, H_2 (most stable of all organic compounds)
Alcohols	H_2, hydrocarbons, glycol, oxidation products, polymer
Carboxylic acids	CO_2, CO, H_2, hydrocarbons
Amines	Lower amines, hydrocarbons, NH_3
Halides	HCl, HBr, I_2, Br_2, dimers
Esters	Acid, alcohol, hydrogen, hydrocarbons
Ethers	H_2, polymer, hydrocarbons
Dyes	Bleaching, oxidation or reduction

TABLE 17

RELATIVE RADIATION RESISTANCE OF SOME ELASTOMERS AND FLEXIBLE PLASTICS IN ORDER OF DECREASING RESISTANCE

(top to bottom)

E L A S T O M E R S
Urethanes
Styrene-butadienes
Natural Rubber Types
Vinylpyridines
Acrylics
Acrylonitrile-butadienes Chlorosulfonated Polyethylenes Neoprenes
Silicones
Fluoroelastomers Polysulfides
Butyl Rubbers

P L A S T I C S
Urethanes
Polyethylenes
Vinyl Chlorides
Fluorocarbons

TABLE 18

RELATIVE RADIATION RESISTANCE OF SOME RIGID PLASTICS IN ORDER OF DECREASING RESISTANCE

(top to bottom)

Polystyrene
Resin Glass Laminates (Phenolic, Silicone, etc.)

Polyester
Polycarbonate

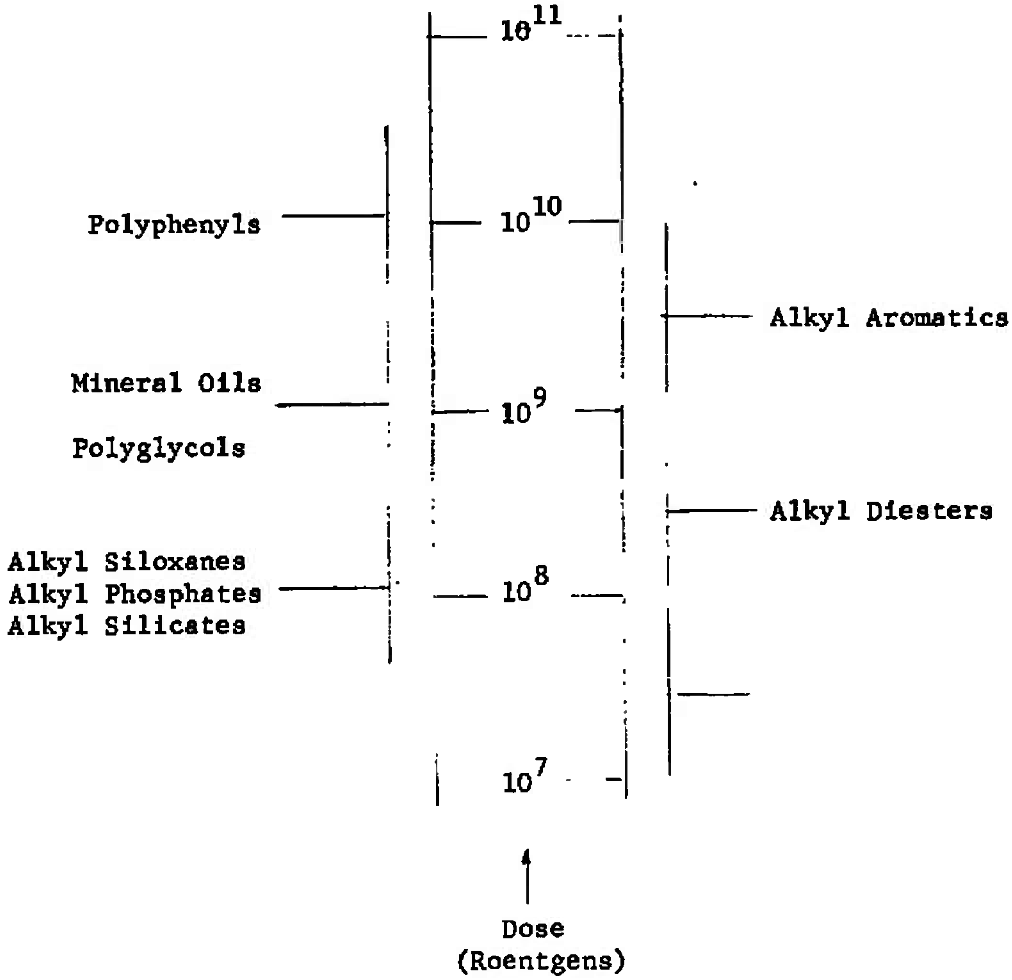
Polyamide

Methyl Methacrylate

Cellulosics

Polyformaldehydes

TABLE 19



RADIATION RESISTANCE OF ORGANIC FLUIDS

Range From Appreciable Viscosity Change
to Solidification is Shown For Each Fluid

SECTION V – BEARING AND GEAR MATERIALS FOR SIMULATOR APPLICATIONS

A. BEARING MATERIALS

The bearing materials used in the program consisted of "off-the-shelf" types. The primary considerations that led to their use are described as follows:

1. Realistic availability of bearings to meet a variety of simulator requirements associated with the various types and sizes of bearings that will be needed.

2. No apparent problems associated with operation in vacuum.

3. Economical, from the standpoint of the costs involved in the more exotic type of bearing material which actually does not appear to merit consideration at this time.

4. Voluminous performance data pertinent to the life and operating characteristics of 52100 bearings with conventional, and some unconventional lubricants is available to draw upon.

A summary of the test bearing types and their respective materials that were used throughout the program are summarized in Table 20.

Perhaps the major justification for going to the more expensive types in the near future would hinge on: (a) need for bearings with increased load carrying capacity, and (b) need for bearing materials that can withstand high lubricant processing temperatures without being tempered back.

Because of anticipated requirements in the near future, therefore, Tables 21 and 22*present a compilation of Rolling Element Bearing Materials Most Commonly Used, and Rolling Element Bearing Materials For Special Use, respectively.

1. <u>Cryogenic Bearing Material Considerations</u>

The problems associated with gear and bearing metals per se are not so much linked with vacuum conditions as they are with cryogenic temperature effects. Fortunately, the cryogenic requirements are not of paramount importance at this time because of the fact that simulator application requirements will be limited to static load conditions (possibly) as described in the footnotes of Table 1, Section I. On the other hand, some consideration is merited at this time because of this possibility, as well as the possibility of future changing simulator operating requirements.

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*Encyclopedia of Chemical Technology, Vol. 3, "Bearing Materials", by R.E. Lee, Jr., E.R. Booser, and D.F. Wilcock, 1964. Published by John Wiley and Sons, New York City.

TABLE 20

TEST BEARING TYPES AND MATERIAL COMPOSITIONS USED IN THE SMALL SCALE AND LARGE SCALE TESTS

Mfgr.	Bearing No.	Type	Type of Loads	Bore Size (MM)	Element Material O.R.	Element Material I.R.	Element Material Rolling Element	Element Material Cage	Element Material End Ring	Internal Clearance (Inches)	Program Use
SKF	22220CY	Spherical	Radial and Thrust	100	52100	52100	52100	Bronze	--	0.003-0.0041	Large Scale Test Bearing
Rollway	EB5220B	Cylindrical	Radial	100	52100	52100	52100	Segments SAE 1110 Ends SAE 1020	Snap Ring SAE 1060	0.0035-0.0051	"
Norma Hoffmann	6220ZZ (220)PP	Ball	Radial	100	52100	52100	52100	SAE 1010	Special Shields[2] Dev.	(3)	"
Rollway	E1206-B[1]	Cylindrical	Radial	30	52100	52100	8620 Case Hardened	SAE 1010	--	(3)	Small Scale Test Bearing
Norma Hoffmann	22206HL[1]	Spherical	Radial and Thrust	30	52100	52100	52100	Brass	--	0.0018-0.0024	"
Micromatic Hone Corp.	Fabroid	Sleeve Bearing	Radial	41.3	PTFE + Glass Fiber Composition			--	--	0.006 diam.	"
SKF	22210C	Spherical	Radial and Thrust	50	52100	52100	52100	Bronze	--	(3)	"
Rollway	E1210B	Cylindrical	Radial	50	52100	52100	52100	SAE 1010	--	0.0022-0.003	"
Timken	Cone 365 Cup 363	Tapered Roller	Radial Thrust	50	52100	52100	52100	--	--	(3)	"

Footnotes:
(1) These bearings were also used for support bearings in the 100 MM bearing test apparatus.

(2) The special shield consisted of an SAE 1010 ring to which was molded Buna N elastomer. This provided an easy to install and disassemble shield for bearing lubrication purposes as well as a good sealing design to minimize lubricant creepage.

(3) The clearance specified for these bearings was C-3, or larger, which required a radial bearing clearance greater than the normal clearance stipulated by the bearing manufacturer. Liberal bearing clearances are necessary where solid film lubricants are to be applied.

TABLE 21*

ROLLING ELEMENT BEARING MATERIALS FOR SPECIAL USE

Bearing material, races and rolling elements	Nominal chemical composition, wt %														
	C	Mn	P	S	Si	Ni	Cr	Mo	V	W	Fe	Co	Al	Cu	Other
MHT	1.0	0.35	0.025	0.025	0.40		1.5				bal		1.0		
M1 Tool Steel	0.80	0.26	0.021	0.03	0.26		3.82	8.5	1.16	1.63	bal				
M2 Tool Steel	0.83	0.27	0.03	0.04	0.30		4.5	5.0	1.00	6.40	bal				
M10 Tool Steel	0.85	0.25			0.30		4.0	8.0	2.0		bal				
M50 Tool Steel	0.80	0.20			0.25		4.10	4.25	1.1		bal				
MV-1 Tool Steel	0.80	0.30			0.25		4.10	4.25	1.10		bal				
Halmo (VM)	0.58	0.27			1.15		4.76	5.25	0.55		bal				
T1 Tool Steel	0.70	0.30			0.30		4.0		1.0	18.0	bal				
Stellite 3	2.45	1.0			1.0	3.0	30.5			12.5	3.0	bal			2.0
Stellite 19	1.7	1.0			1.0	3.0	31.0			10.5	3.0	bal			2.0
Stellite 25	0.10	1.5			1.0	10.0	20.0			15.0	3.0	bal			
Stellite Star J	2.50*	1.0*			1.0*	2.50*	32.0			17.0	3.0*	bal			2.0
K Monel						66.0					0.9		2.75	29.0	

* Maximum amount.

TABLE 22*

ROLLING ELEMENT BEARING MATERIALS MOST COMMONLY USED

Bearing materials, races and rolling elements	Nominal chemical composition, wt %									
	C	Mn	P*	S*	Si	Ni	Cr	Mo	V	Fe
AISI 3115	0.10	0.45	0.04	0.05	0.20	1.25	0.60			balance
AISI 4320	0.20	0.55	0.04	0.04	0.26	1.82	0.50	0.25		balance
AISI 4615	0.15	0.53	0.04	0.05	0.22	1.82		0.25		balance
AISI 4820	0.22	0.60	0.04	0.04	0.27	3.50		0.25		balance
AISI 5120	0.20	0.45	0.04	0.05	0.22		0.75			balance
AISI 6120	0.20	0.45	0.04	0.05	0.22		0.95		0.16	balance
AISI 6195	0.98	0.32	0.03	0.035	0.22		0.95		0.16	balance
AISI 8620	0.20	0.80	0.04	0.04	0.27	0.55	0.50	0.20		balance
AISI 52100	1.0	0.35	0.025	0.025	0.25		1.5			balance
AISI 420	.30	1.0*	0.04	0.04	1.0		13.0			balance
AISI-440C	1.0	1.0	0.04	0.04	1.0		17.0			balance

* Maximum amount.

*Encyclopedia or Chemical Technology, Vol. 3, "Bearing Materials", by R.E. Lee, Jr., E.R. Booser, and D.F. Wilcock, 1964. Published by John Wiley and Sons, New York City.

Cryogenic temperatures can be expected to increase the mechanical properties of most ferrous and non-ferrous metals in general with the exception of the impact strength property. A significant reduction of the impact strength property of a material will subject the material to brittle fracture under shock loads. Physical structure has an important effect on ductile or brittle behavior. Face-centered cubic metals such as copper and the stable austenitic steels (300 series) remain ductile at low temperatures while body-centered cubic metals such as carbon steels and hexagonal-close packed metals like zinc generally have a transition from ductile to brittle behavior at temperatures near room temperature.

Fortunately, some work has been carried out on rolling element bearings operating in low temperature environments while little gear information is available.

The following discussion concerns the structural characteristics of the bearing materials. As in the case with gear materials operating at low temperatures, impact strength is a major consideration. The austenitic stainless steels, most of the aluminum and copper alloys retain sufficient ductility at low temperatures. AISI 303 stainless steel for example has over twice the tensile strength at minus 423° F that it has at room temperature. The notch toughness decreases with decreasing temperature but the impact strength is still reasonable. The tensile strength of AISI 1010 increases to about three times the room temperature value while the impact strength drops from about 15 ft-lbs. to 2-3 ft-lbs. This would indicate that ordinary carbon steel is extremely marginal for low temperature use. At minus 320° F, alloys 17-7 PH and AISI 4340[3] indicate a considerable ductility retention, as measured by elongation and reduction of area, of 80 percent of their room temperature values.

AISI 52100 and AISI 440C have been under investigation for use as low temperature rolling contact bearings and the results to date look promising although some limitations have been noted.

In tests conducted by the National Bureau of Standards,(1) standard "off-the-shelf" AISI 52100 series steel single row, radial bearings with pressed steel separators were evaluated at cryogenic temperature (-321° F liq. N_2) in a pump testing device. These bearings gave only limited service because of mechanical failures occurring in the separators. One type of failure occurred with a clipped pressed steel separator. Fatigue of the clips at points of high stress concentration resulted in the failure of the metallic separator. No wear was evidenced on the balls or races.

Going to a riveted metal separator resulted in severe separator wear and subsequent failure. Resorting to fabric-reinforced phenolic separators in standard 52100 series angular contact bearings resulted in satisfactory performance under the prescribed test conditions. It was observed in this work that careful handling of the bearing material as well as the prevention of moisture accumulation was necessary to prevent rusting.

A 440C bearing was later selected for test [2] because of its ability to be hardened, resistance to wear, and improved corrosion resistance over that of 52100. Again evaluating various types of separators it was found that a filled PTFE separator riding the outer ring of a 440C bearing resulted in satisfactory performance under the condition of the prescribed test. It was also observed that 52100 steel may have a slight advantage over 440C stainless steel although it was concluded that 440C steel should be used for ball and race materials of bearings in simulators where corrosive environment may be present. Scibbe and Anderson [3] in recent ball bearing performance studies in liquid hydrogen used ball and race materials of either AISI 52100 or AISI 440C in conjunction with glass-fiber-filled PTFE separators with quite successful results.

These results would indicate that both AISI 52100 and 440C are suitable materials for rolling element bearings operating in cryogenic environments.

AISI 4340 is an attractive possibility for the gear material because of its strength and durability at temperatures to minus 320° F.

References for Cryogenic Bearing Material Considerations

1. Martin, K.B., Jacobs, R.B., "Testing and Operation of Ball Bearings Submerged in Liquified Gases". ASLE Paper 58LC13, Oct. 1958.

2. Wilson, W., Martin, K., Brennan, J., and Birmingham, B., "Evaluation of Ball Bearing Separator Materials Operating Submerged in Liquid Nitrogen". ASLE Paper No. 60LC-4, 1960.

3. Scibbe, H.W., Anderson, W.J., "Evaluation of Ball Bearing Performance in Liquid Hydrogen at DN Values to 1.6 Million". ASLE Trans., Vol. 5, No. 1, April, 1962.

B. GEAR MATERIALS

The primary function of a gear is to transmit power. To perform this function properly the gear material must possess adequate load carrying capability, and be resistant to wear, pitting and fatigue. Design variables and properties pertinent to meeting these requirements consist of the following:

1. Strength
   a. Gear Material
   b. Configurations

2. Toughness
   a. Hardness
   b. Impact (especially at cryogenic temperatures)

3. Wear Resistance
   a. Hardness
   b. Strength
   c. Compatibility

4. Fatigue Resistance
   a. Surface Finish
   b. Gear Material Quality (cleanliness)
   c. Compressive Strength

An attempt was made to use "off-the-shelf" materials for the test gears in this program. Unfortunately, while reasonable performance resulted with some of the lubricants investigated, the gears should have been considerably harder. The additional expense for gear materials with higher hardness characteristics is justified from the standpoint of improved lubricant film and gear material life.

Small and Large Scale Gear Materials

	Material	
Test Use	1-1/4" PD Helical Pinion	4" P.D. Helical
Small Scale Tests	AISI 4140, (Rc 39)	Meehenite
Large Scale Tests	AISI 4140, (Rc 39)	AISI 4340, (Rc 42)

Note: As an example of cost, an AISI 4340 helical gear with a 4" PD and hardened runs in the neighborhood of $46.00. The 4" PD Meehenite gear ran $16.00.

A careful selection of representative gear materials, most of which are used throughout industry today, is presented in Table 23. The group at the top of the table were selected on the basis of their high strength properties which are indicative of high load carrying capability. The two non-ferrous materials at the bottom of the table have considerably lower strength characteristics but on the other hand possess generally good impact properties at low temperatures.

TABLE 23

PROPERTIES OF POTENTIAL GEAR MATERIALS

Gear Material	Brinell Hardness	Tensile Str. (psi)	Yield Str. (psi)	Elong. In. 2 in. (%)	Red. of Area (%)	Impact Str. ft-lb	Remarks
Nitralloy 135 Modified	310	138,000	110,000	4	17	44	
AISI 4340	475	248,000	216,000	13	49	13	Good ductility down to -320°F
AISI 4640	475	243,000	227,000	12	47	13	
AISI 3310	375	181,500	149,000	15	57	40	
AISI 410	375	180,000	140,000	15	--	35	
AISI 4140	475	240,000	220,000	12	47	9	
Nickel Cast iron	235						
Meehanite Type GA	207	50,000					
Ductile iron 60-45-10	140/190	60,000/80,000	45,000/60,000	10/25		60/115	
Gray Iron Class 60	252	62,500	187,500				cryogenic behavior not considered good
Phosphor Bronze A	160	68,000	55,000	28			copper base alloys-generally good impact prop. at low temp.
Nickel Bronze Ni-Vee A	180	85,000	55,000	10	26	110	same as above

Notes: 1. Most of the above materials are considered outstanding for gear components at room temperature and above. Specific information on their performance at low temperature is not available.

2. Optimum hardnesses may be obtained by appropriate heat treatment pertinent to that material. Examples are: through hardening, nitriding, carburizing, induction and flame hardening.

SECTION VI – COATING AND PLATING PROCESSING PROCEDURES

INTRODUCTION

This section describes in detail how the various bearings and gears were prepared and subsequently processed according to the coating or plating used in the small and large scale space simulator tests described in Sections IX and X.

A. BEARING AND GEAR PREPARATION PRIOR TO PROCESSING

1. Bearings were disassembled so that cage, inner and outer race, and rolling elements were separated. Certain types of bearings like the 2220CY spherical bearings for example, can be disassembled by the user. Other types, like the EB5220B cylindrical roller bearings, have to be ordered disassembled from the bearing vendor, coated or plated by the user, and returned to the vendor for reassembly.

Gears should also be separated from their respective housings for coating or plating purposes.

2. Clean according to "Cleaning Procedure For Gear and Bearing Elements", described in Table 24 of this section. Follow steps 1-9 when a thin film plating is to be applied. An additional step consisting of vapor blasting was carried out between steps 2 and 3 when lubricant coatings containing binders were to be applied.

A reasonably good vapor blasting, easily observable to the naked eye, is necessary to insure a good coating adherence. The rolling elements in this program were not vapor blasted, or plated or coated, because:

a) It is felt that the solid solubility tendency of the same plating in contact with itself increases the "cold welding" potential.

b) Loose bearing operating clearances are necessary, after the races and retainer have been plated, or coated, to minimize any jamming potential resulting from the generation of wear debris over a given operating period, as well as allowing for sufficient running clearance.

   Hence, the balls and roller elements were not vapor blasted but cleaned according to steps 1-9 of Table 24, with one exception. In place of cathodical cleaning (step 5), each element was cleaned with wet levigated alumina (15.0 micron size).

c) Judgment should be used as to vapor blasting or cleaning with levigated alumina retainer or cage elements prior to coating. Entrapment of abrasive particles on the surface of a sintered type of retainer structure as well as other material types that are porous, or contain surface irregularities as a characteristic part of their microstructure, could occur, thereby resulting in reduction of coating effectiveness.

d) After cleaning use protective gloves in handling parts for balance of processing steps, as thumb and finger prints on the clean metal surfaces can detrimentally affect coating adherence. As a matter of fact one can etch his thumb print on a clean AISI 52100 bearing steel surface quite easily.

A description of the cleaning procedures, as well as plating and coating processing techniques, follows.

B. CLEANING PROCEDURE FOR GEAR AND BEARING ELEMENTS

TABLE 24
GENERAL CLEANING PROCEDURE

The test bearings and gears are cleaned prior to coating, or plating, in the following described manner. Some of these steps are flexible depending on the parts to be cleaned as previously indicated.

Steps

1. To remove packing grease brush thoroughly in benzene (1) under protective hood.

2. Use levigated alumina (wet, on soft clean cloth) on races and rollers if noticeable stain is observed on new bearings or glass after removal of grease. Entire race surface must be thoroughly cleaned to prevent migration of contaminent.

3. Repeat step 1, if 2 is carried out.

4. Rinse in clean benzene under protective hood.

5. Cathodically clean gear or bearing elements in an 80° C (176° F) bath of 10% trisodium phosphate, 90% distilled water (by weight). Using a palladium anode (+) (2), attach part to be cleaned to the negative terminal (-) of a 22-1/2 volt battery, submerging the part to be cleaned in the bath for 30-45 seconds.

6. Immediately rinse in warm tap water for 15-20 seconds. Good "wetting" of water on metal surface is rough indication of how well surfaces were cleaned.

---

(1) Toluene may be substituted for benzene as the latter should be used under a hood because of its proported toxicity.

(2) The use of a steel anode resulted in plating or depositing an undesirable film on the part to be cleaned. We subsequently replaced the steel anode with palladium and this problem was minimized.

7. Immediately after step 6 rinse in reagent type alcohol, preferably by squirting fresh alcohol over all element surfaces.

8. Air dry.

9. Immediately apply coating, or place in clean, inert environment until ready for plating.

Steps 1 to 9 prepare the elements for plating. In addition, further cleaning is conducted by the processor prior to plating, as indicated under Platings (VI-C).

When lubricant-binder types of coatings are to be used on the surface of an element, a vapor blasting treatment should be carried out between steps 2 and 3. The vapor blasting treatment consists of a slurry of silicon oxide (320 mesh) plus water, sprayed through an atomized nozzle under a pressure of 50-60 lbs. Air agitation of the slurry keeps the particles in suspension throughout spraying.

C. PROCESSING

Platings

The plating procedures are described as follows:

23 Kt Gold Plate on AISI 52100 Steel Races

1. All parts vapor degreased.

2. Parts electrocleaned in 180° F Oakite (alkaline) solution.

3. Rinse several times in hot clean water.

4. Acid etch (approx. 30 sec.) in inhibited muriatic acid.

5. Double water rinse with a subsequent dilute sodium cyanide dip (neutralizer).

6. Rinse.

7. Copper flash in copper cyanide bath (1) - flash thickness estimated at 1×10^{-6} inches thick.

8. Double water rinse.

9. Parts are then dilute-acid dipped (pre-plating neutralizer).

---

(1) Rochelle copper high speed bath.

10. Double water rinse.

11. Parts are then 23 kt gold plated (1) to desired thickness. (Current on when entering tank.)

12. After plating, parts are dipped in alkaline neutralizer to offset acidity of gold bath.

13. Rinse, and immerse in reagent alcohol.

14. Air dry.

For different retainer materials, the procedure is the same as above with the following exceptions:

Cast iron retainer - shorten acid etch time (step 4) to 15 - 20 seconds.

Bronze retainers - additional acid etch (step 4) of 10 percent Fluoboric acid (15 - 30 seconds).

24 Kt Gold Plate

Same as the 23 kt gold plate procedure with the exception of plating, which consists of Temperex HD (24 kt gold) made by the Sel Rex Corporation, Nutley, N.J.

Silver Plate

Same as 23 kt gold procedure with the following exceptions:

Omit step 9, i.e., dilute-acid dip (pre-plating neutralizer), and proceed to silver plating step.

Silver plate consists of Sel Rex Industrial Silver process "Silvrex".

Plating Technique

Parts to be plated with gold or silver enter the tanks with the current on. Gold plate is applied at 100^{o} F, pH of 3.5 electromatic, 1 troy oz/gal. gold, S.G. of 9.5 Be with vigorous agitation. 0.0001 inch of gold plate is deposited in 14.3 minutes at a current density of 10 amp/ft^2.

Silver plate comes from a working solution of 10 oz/gal silver metal at R.T., pH = 12.3 approx., with vigorous agitation. 0.0001 inch of silver is deposited in 1.85 minutes at a current density of 20 amp/ft^2.

(1) 23 kt gold plate (Sel Rex Karatclad 368 Series) contains alloying additives of 5 gm/liter nickel and 1 gm/liter of indium.

Plating Thickness Measurements

Plating thicknesses are double-checked with the Lea Manufacturing Corp. "Electromag", magnetic thickness tester (using National Bureau of Standards standards), against the time/current density computation.

Coatings

MoS_2 + Graphite + Sodium Silicate Binder Coating

Composition: 71.0% MoS_2
7.0% Graphite
22.0% Sodium Silicate (Na_2O ratio to SiO_2) = 1:2.9

Commercially available as Alpha MolyKote X-15

1. Follow the steps, where applicable, outlined in Table 24, "General Cleaning Procedure.
2. After insuring that the lubricant has been well mixed in its container, carefully brush the liquid coating on the surface of bearing element or gear teeth to be coated. Spraying may also be considered; however, brushing was used because parts inaccessible to direct spraying can be reached easily with a brush. The coating goes on more uniformly when applied at a reasonably fast rate, as coating drips quite rapidly. An increase in coating uniformity was accomplished by slowly rolling the outer race and allowing excess liquid lubricant from the brushing to flow over the uncoated surfaces.
3. Allow 30 minutes air dry in a clean area.
4. Bake in oven at 180° F for one hour.
5. Buff down excess coating from raceways.
6. Assemble and "pre-run-in" for approximately 25 cycles at 2 - 6 rpm, in air, prior to operating in vacuum.

MoS_2 + Epoxy Binder Coating

Commercially available as Alpha MolyKote X-106.

1. Follow the steps, where applicable, outlined in Table 24, "General Cleaning Procedure."
2. Brush or spray on well-mixed lubricant.
3. Air dry in clean environment for 10 minutes.

4. Bake in oven at 300° F for 60 minutes.

5. Buff down excess coating from raceways.

6. Assemble and "pre-run-in" in air for approximately 25 cycles at 2-6 rpm, prior to operating in vacuum.

MoS_2 + Glass Binder

ATL developmental coating.

1. Clean gears and bearing elements according to steps outlined in Table 24, "General Cleaning Procedure".

2. The procedure used to make the lubricant is described in Section III-B, "Inorganic Lubricant Development". At the present time the lubricant is in the form of a composite "stick".

3. The method of applying the lubricant to date has been to "rub" it on the surface to be coated. This can be accomplished by rubbing it on by hand, that is, holding it in a small fixture that can be hand-held, with the element to be coated stationary.

   Another method which was used, was to carefully hold the inner and outer races in a lathe chuck, turn the lathe at around 200-300 rpm, and hold the lubricant composite stick against the surface to be coated. While the latter method worked out quite well, there is room for improved methods.

4. After a thin observable coating of the MoS_2 - glass lubricant has been applied, the elements should then be placed in an oven where the coating is sintered at 190° C for ten minutes. This time may be varied depending upon the size of the bearing elements or gears associated with sufficient heating, to result in sintering of the coating.

Greases

Methyl Chlorophenyl Silicone

Commercially available as Versilube G-300 Grease.

G-300 is a General Electric grease consisting of methyl chlorophenyl silicone with a lithium soap thickener. Several of its pertinent properties are listed as follows:

Typical Properties of Versilube G-300(*)

Service Temperature Range	-100 to +450°F
Bleed (100 hrs at 302°F)	3% max.
Evaporation (50 hrs at 302°F)	2% max.
Low Temperature Torque 1 rpm, starting torque = 1000 gm	-94°F

1. Follow steps where applicable as outlined under "Cleaning Procedure for Gear and Bearing Elements".
2. Apply grease with wooden spatula, working it in carefully and thoroughly to surfaces of bearings or gears.

   While there is no hard and fast rule for the quantity of grease lubricant to be used in a bearing, it is recommended that no more than 50% by volume be filled, based on the space available between inner and outer race of a typical rolling element bearing. The same quantity appears practical for gears in space environment usage.

Petroleum Oil Distillate

Commercially available as Apiezon L. Apiezon L is a petroleum base distillate available from James G. Biddle Company, Pennsylvania, and the Shell Oil Company. Several of its pertinent properties are listed as follows:

(*) Versilube Silicone Lubricants, Technical Data Book S-10, Silicone Products Dept., G.E.Co., Waterford, N.Y.

Typical Properties of Apiezon L

Service Temperature	86°F max.
Temperature for Application	room temperature
Melting Point	116.6°F
Vapor Pressure (mm of Hg) after evolution of dissolved air	10^{-10} to 10^{-11} at room temperature

Apiezon L was successfully run over a limited test period in the 50 mm and 100 mm bearing tests reported in Sections IX and X. A specially designed snap shield was used to contain the lubricant in the 100 mm bearing test. The shield consisted of an AISI 1010 steel frame with a Buna N (elastomer) bead attached. This allowed for:

1. Good retention of grease in bearing.
2. Convenient snap-out feature, allowing ease of initial and subsequent filling of bearing with lubricant.

Apply grease as per items 1 and 2 described under "Methyl Chlorophenyl Silicone".

D. EFFECT OF FILM PROCESSING TEMPERATURES ON BEARING STEELS

The test bearings for all of the aerospace chamber tests consisted of AISI 52100 steel. Within its limitations it is an excellent bearing material and can be obtained as an "off-the-shelf" item quite readily as well as economically.

In applying some types of thin films care must be exercised as to the choice of baking, curing, or lubricant sintering temperature. When PTFE was considered as a thin film lubricant, for example, it was noted that fusing temperatures could range in the order of 650-750° F. This results in a considerable softening of the AISI 52100 steel, for example, which reduces bearing properties and adversely affects film life. The extent of softening from the fusing temperature (which acts as a tempering treatment) can be seen in Table 25.

No known work to date has indicated a method of adequately bonding films of PTFE to a metal surface without a sintering operation. Air dried coatings are removed quite easily because of lack of good bonding. Consideration has been given to radiation techniques for PTFE bonding along with proper associated metallic fillers; however, the feasibility of this technique has not been proven practical. To circumvent this problem, a bearing material would have to be selected that would not be metallurgically affected by the fusing temperature. One can immediately see in Table 25 that AISI 440C is also considerably reduced in hardness after a 600° F temper, whereas M2 and M50 tool steels maintain satisfactory hardness levels at tempering temperatures whose resulting hardness would not be affected by a high fusing temperature. It was concluded from this that it was not feasible to apply a PTFE coating to the 52100 test bearings and further, that in the selection of lubricant films in general, where a processing or curing temperature is required, careful consideration must be given as to the effect of the processing temperature on the mechanical properties of the metal to be coated.

TABLE 25

TEMPERING TEMPERATURE VS. HARDNESS FOR SEVERAL BEARING STEELS

Bearing Steel	Tempering Temperature	Resulting Hardness (Rc)
AISI 52100	As Quenched	65
	300	63
	400	62
	500	59
	600	57
	700	53
	800	49
AISI 440C	As Quenched	61
	300	60
	400	57
	500	56
	600	56
	800	56
M2 Tool Steel	As Quenched	65
	400	63
	600	63
	800	60
M50 Tool Steel	As Quenched	64
	900	61
	1000	63
	1025	64
	1050	63
	1100	60
	1200	47

E. PLATING AND COATING THICKNESSES USED IN THE SMALL AND LARGE SCALE BEARING AND GEAR TESTS

Lubricant	Thickness-In. (approx.)
24 kt Gold	0.0001 - 0.0002
23 kt Gold	0.0001 - 0.0002
Silver	0.0001 - 0.0002
MoS_2 + Graphite + Silicate Binder	0.0003 - 0.0005
MoS_2 + Epoxy Binder	0.0003 - 0.0005
MoS_2 + Glass Binder (ATL)	0.0003 - 0.0005

A copper strike was used in conjunction with these platings (metal) for the following purposes:

1. Improve plating adherence.
2. Improve corrosion resistance of base metal.
3. Possibly improve load carrying capability of plating.

SECTION VII - EVALUATION TECHNIQUES

A. BEARINGS

One of the most important foundations for setting up an evaluation program for bearings is the determination of suitable scaling and acceleration factors for testing.

In most rolling element bearing applications, life is predicted on the basis of a statistical fatigue correlation. Load and rotational speed are the imposed factors which determine life. This assumes, however, that the bearings are adequately lubricated according to some predetermined standard. Even in these more or less conventional applications, bearing failure may occur -- because of lubricant depletion or degradation -- before the bearing materials actually reach their fatigue limit.

In the simulator applications dry film lubrication is desired. In addition, the applications consist mostly of low speed rotation or oscillation and relatively heavy loads. Previous experience in applications of this type shows that the failure mechanism is not fatigue, but rather a failure by intolerable wear and/or friction torque, or a gross catastrophic failure such as cage or retainer breakage.

Many long duration tests of large bearings are impractical from the standpoint of time and cost. It was therefore desirable to scale up the performance of smaller, more easily evaluated bearings, and to be able to extrapolate the results of short time tests to predict performance over a longer design life period. Thus the need for scaling and acceleration factors.

A scaling factor was therefore employed based on static capacity of a rolling element bearing or, synonomously, running smaller bearings at about the same Hertz stress as would be experienced by larger bearings. See Table 26 for load vs. Hertz stress for 30 mm rolling element bearings.

Because of its simplicity, the ratio of bearing capacity to applied load was the most desirable scaling factor. The testing of several bearing types was necessitated for differing functional requirements of the applications and relative adaptability to dry film lubrication. Different bearing types and a variety of load and speed design conditions cause varying degrees of sliding between bearing parts -- which in itself has a significant effect on film life.

The introduction of a dry film lubricant between a rolling element and a race alters the compressive stress distribution only slightly in the applications under study. This conclusion is partially based on the analytical work of Dowson and Higginson ("Effect of Materials Properties on the Lubrication of Elastic Rollers", Journal of Mechanical Engineering Science, Vol. 2, No. 3, Sept., 1960, pp. 188-194) which shows that, for low speeds, the Hertz stress distribution in the base metal is practically identical whether an incompressible lubricant is present or not.

TABLE 26

ATL SMALL SCALE BEARING TEST LOADS

(30 mm Rolling Element Bearings)

BEARING TYPE	COMPRESSIVE STRESS (MEAN)	LOAD RQD (PER BRG)
E1206B Rollway Cylindrical	200,000	1715
	190,000	1550
	180,000	1395
	170,000	1240
22206 HL Norma Hoffman Spherical	200,000	3670
	190,000	3300
	180,000	2980
	170,000	2670
HJ-243316 Torrington Needle	200,000	4730
	190,000	4270
	180,000	3850
	170,000	3420
HDR-206 Split Ball	200,000	252
	190,000	215
	180,000	183
	170,000	155

Based on the work represented by the previous discussion, it was concluded that the ratio of bearing static capacity to applied load was a reasonable scaling relationship, i.e., C_o/P.

The selection of acceleration factors for testing was a more complex problem. There are four basic environmental variables to the bearing which can accelerate failure. These are load, rotational speed, ambient temperature, and ambient pressure. Ambient pressure was fixed at a minimum value to reproduce expected simulator conditions; it allowed, therefore, practically no usefulness as an accelerating factor. Ambient temperature could have been made much higher than that nominally expected in the applications. This, however, would probably unrealistically affect the performance of lubricant films. This narrowed the possibilities to load and rotational speed. Since the imposed loads were already close to the allowable limits of the materials, there was a very narrow load range available for accelerating failures.

By this rationalizing process of elimination, then, rotational speed was left as the accelerating factor. Test speed was limited, however, to some maximum value determined by a judgment as to where the lubricating mechanism of the film might change and where the kinematic and dynamic characteristics of the bearing would be seriously different from the low speed conditions of the applications.

Scaling Factors

Rolling Contact Bearings

A scaling factor was introduced so that small scale testing produced meaningful data for full scale extrapolation. Essentially, it consisted of subjecting a small rolling element bearing to the same compressive stress that will be encountered in the actual application. The test bearing geometry, environment, dry film lubricant, and stresses were the same as that of a full scale bearing; hence an approximation of frictional torque, film life, and general operating behavior could be predicted for the actual application.

The radial load on a cylindrical type roller can be related to a local stress in the bearing as follows:

$P=K\,(S)^2$ where P = applied radial load

K is a function of the bearing geometry, size, material properties, clearance.

S = mean compressive stress of the most heavily loaded ball or roller

For the same bearing the static capacity may be expressed as:

$C_o = K_b\,(S_o)^2$ where C_o is the specific static capacity as determined by the bearing manufacture

K_b is a constant for the given bearing

S_o is the mean compressive stress which the manufacturer determines to result in a deformation of 0.0001 x the diameter of the most heavily loaded ball or roller.

Therefore for a given bearing

$$\frac{C_o}{P} = \left(\frac{S_o}{S}\right)^2$$

If S_o is known for a type of bearing from a given manufacturer, then S may be rapidly computed

$$S = S_o \left(\frac{P}{C_o}\right)^{1/2},$$

this allows running small and large bearings (of the same type and source) at the same stress level.

E.g., if S_o = 500,000 psi and $C_o/P = 3$ for a "full size" bearing, then S =

$$500{,}000\ (1/3)^{1/2} = 289{,}000 \text{ psi.}$$

Further -- if C_o is known for the small bearing, then the required P for small scale testing can be computed.

Journal Bearings

A journal bearing was considered for some of the applications described in Table 1 of Section I. Where self-alignment is required, a self-aligning adapter will have to be incorporated it a journal bearing is to be used. Based upon a length to diameter ratio of 1, the maximum unit loading of a journal bearing that would occur for application No. 2, Section I, Table 1, would be:

$$S = \frac{P}{LD} = \frac{290{,}000 \text{ lb.}}{(12)\ (12)} = 2{,}000 \text{ psi}$$

L = length (inches)

D - Inside Diameter (inches)

For the Drawbridge application, No. 12 in Table 1, for example:

$$S = \frac{5000}{(6)(6)} = 139 \text{ psi}$$

Frictional torque, wear characteristics, and general performance behavior were determined for several different load levels using a PTFE-glass fiber sleeve bearing.

Accelerating Factors

No known data appears to be available on wear rate of a dry film as a function of velocity, for rolling contact bearings. However, pure sliding tests on various materials by R. L. Johnson, M. A. Swikert, and E.E. Bisson indicate a linear relationship (Lubrication Engineering, May-June, 1955). For example, as the velocity of the slider was increased ten-fold, the wear for a given sliding distance was doubled. If this relationship were true for the dry film lubricants selected, then the wear of the dry film lubricant at 1 rpm for 1000 cycles is equivalent to the wear that can be expected at 10 rpm for only 500 cycles. Thus when increasing the speed (accelerating factor), the equivalent film life at the lower speed must be corrected. The correction factor would be:

$$F_C = \frac{1}{9}\frac{(V_T)}{(V_A)} + \frac{8}{9}$$

where: F_C = correction factor

V_T = velocity of accelerated test

V_A = velocity of actual application

The equivalent number of cycles at application speed would then be represented by the following empirical formula:

$$\text{Cycles}_{\text{equiv.}} = F_C V_T \times (\text{time})$$

$$= \left(\frac{V_T}{9V_A} + \frac{8}{9}\right)(V_T)\ (\text{time})$$

Preliminary small scale tests of film life vs. sliding velocity were conducted in conjunction with screening of lubricants.

Small Scale Bearing Test Apparatus Limitations

The size of the bearings to be tested was limited by the test equipment capability. A magnetic coupling capable of transmitting 42 lb-in. of torque in the ATL vacuum chamber was available. The allowable frictional torque generated in the test bearings was therefore limited to 42 lb-in. Since bearing stress level can readily be determined, the bearing size can be calculated. An estimate of the torque required was as follows:

$$T = 4\ Fr = 4fPr$$

where:
T = torque
4 = number of bearings
F = friction force per bearing
f = coefficient of friction
P = bearing radial load
r = pitch radius of bearing

The maximum bearing loading for Category I loads is approximately one-third the static capacity rating, or:

$$P = \frac{C_o}{3}$$

Assuming a coefficient of friction of 0.005, based on previous experience, yields:

$$42 = (4)\quad (0.005)\left(\frac{C_o}{3}\right)(r)$$

$$C_o r \quad = \quad 6{,}375$$

Based on the preceding equations and several bearing geometries, the initial bearing I.D. was established as 1.1811 inches. In later tests however, 50 mm bearings with an I.D. of 1.9685 were run in the small scale bearing tester. Further C_o/P ratios of 3 were found to be too high for the bearings and the loads were reduced accordingly.

A comparison of radial load and corresponding mean compressive stress of several bearing types and sizes considered for small scale testing can be seen in Table 26.

Lubricant Life - Duty Cycle

To provide some information for practical determination of small scale test speeds, duration of test, and a reference or yardstick for approximation of lubricant life, a duty cycle was established. Figure 7 shows graphically the total test revolutions vs. test time at various inner race rotational velocities. This figure may be compared to Figure 8, which shows the total revolutions of a bearing operating at various percentages of time at two revolutions per minute for a one-year period. The 2 rpm was indicative of maximum simulator applications speeds (see Table 1, Section I); however, the

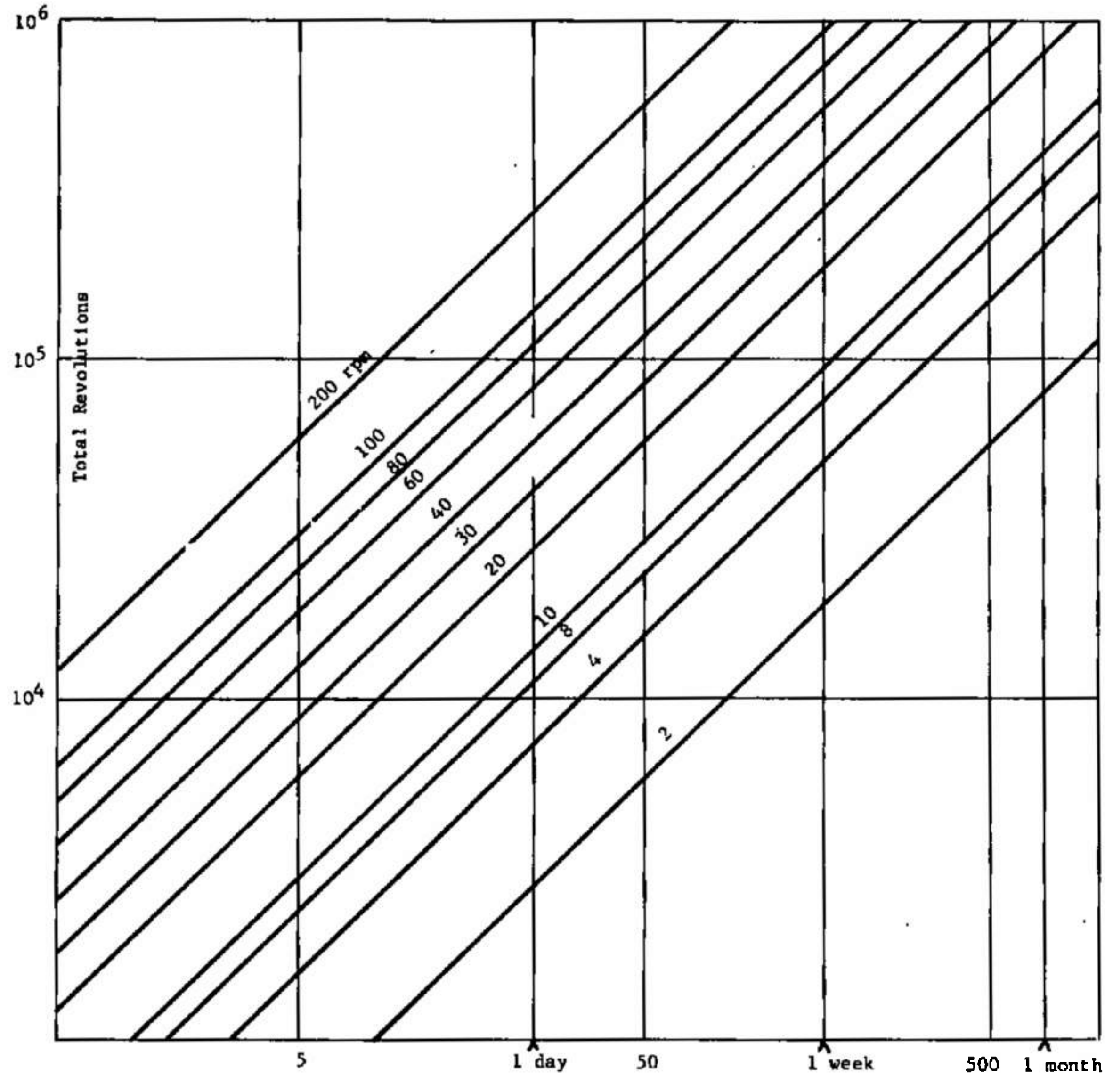

Fig. 7 Total Revolutions versus Test Time at Various Speeds

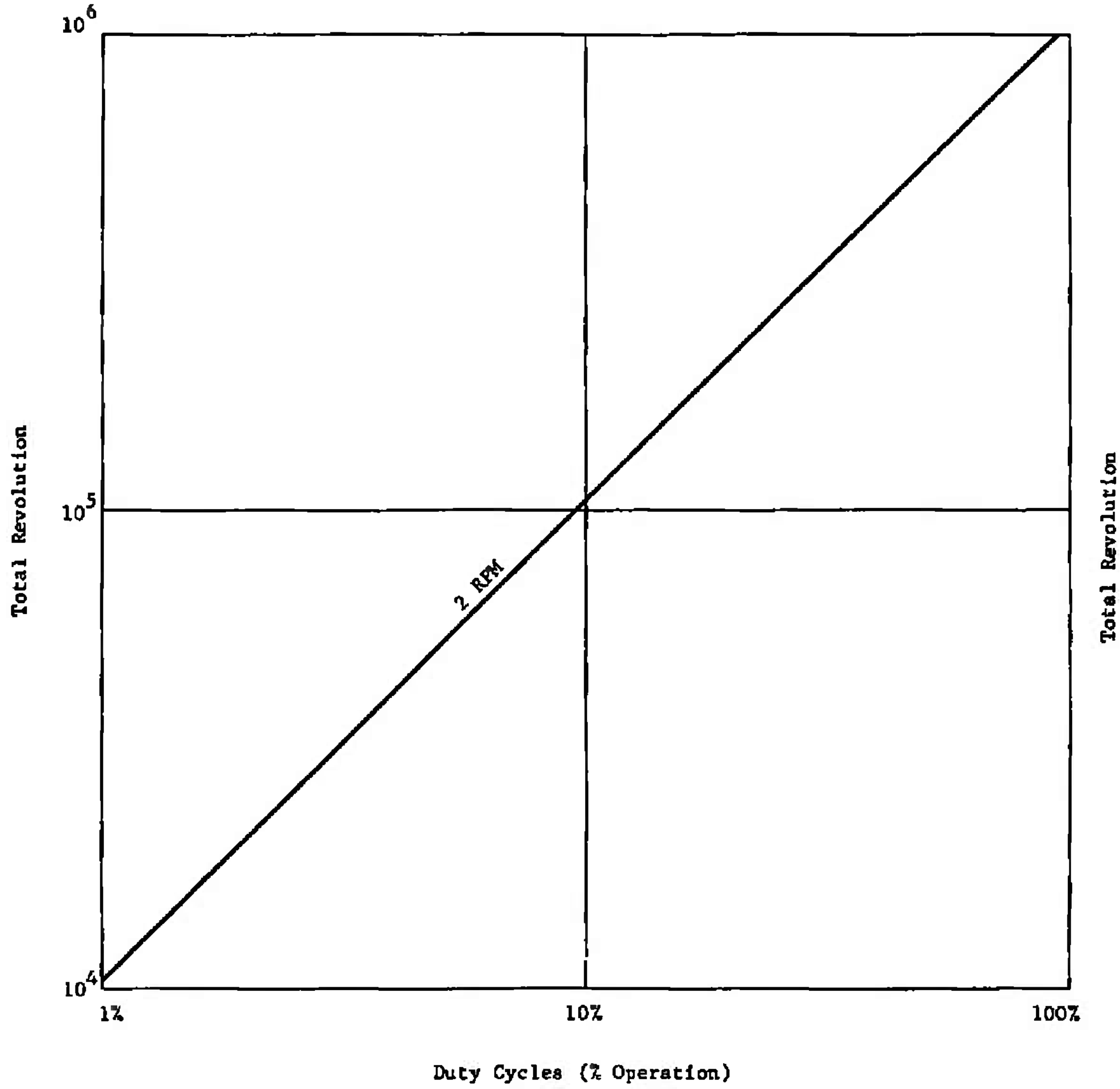

Fig. 8 Total Revolutions at 2 rpm for a One-Year Period at Various Percentages of Time

total of cycles established for a one-year run was for reference purposes only and did not necessarily imply that this was a typical run period for the simulator. Some bearing and gear components, for example, may only be required to operate 25,000-50,000 cycles over a period of one year, some more, others less.

B. GEARS

Full Scale Gear Tests

The setup used to conduct the "Small Scale Gear Tests" utilized test gears of 3.2:1 ratio and of 4 inch center distance. This size was about that actually found in generally available hoisting equipment at the high speed end. Thus these tests closely duplicated the meshing conditions found in motor driven hoists and the high speed gearing in transfer carts.

Gearing at the low speed ends of the equipment (carts and hoists, for example) would be of a somewhat larger size. The loads encountered by these gears would be much heavier in terms of actual force, and would occur for few life cycles.

Although the Small Scale Gear Tests could not duplicate exactly these conditions, due to the much smaller surface radius of the contact portions of the gear teeth, present data indicated that the geometry factors involved could be reduced or enlarged within reasonably wide limits and still achieve good correlation with test results.

It was planned therefore to test several full scale low speed gears under heavy loads similar to those that they would encounter in service to determine the actual degree of correlation between the test performance of these gears, and the performance prognosticated by the small scale tests and the suitable geometry factors.

Scaling Factors

One of the limitations on the load capacity of gears is the ability of the surfaces to withstand the Hertzian contact loads. Evidence of excessive loads appears in the form of pits near or below the pitch line. These surface defects result from the combined effect of sliding and rolling pressures exerted by the meshing teeth on each other. Only at the pitch line (operating) is there no sliding.

In the last century, H. Hertz of Germany established the theory of the stresses in surface rolling on each other. He showed equations expressing the relationship between surface radius of curvature stress and contact area. Also an expression for distance below the surface to the point of maximum shear is shown. In the case of spur gear the area of contact is a rectangular band extending the length of the active surface of the gear tooth parallel to the gear axis. In the case of a helical gear this band becomes an ellipse which lies obliquely along the active surface of the tooth.

The expression for the width of the band of contact (B) in the case of spur gears is

$$K_1 = \frac{1-v_1^{\;2}}{\pi E_1}$$

$$K_2 = \frac{1-V_2^{\;2}}{\pi E_2}$$

$$B = \sqrt{\frac{16F(K_1+K_2)\;R_1R_2}{L(R_1 + R_2)}}$$

where

V_1 & V_2	=	Poisson's ratio of pinion & gear material
E	=	modulus of elasticity
R_1 & R_2	=	radius of curvature of teeth at point of contact, pinion & gear.

Hertz shows that the maximum compressive stress is

$$S_c = \frac{4f}{L\pi B}$$

and the maximum shear stress is

$$S_s = .295\ S_c$$

If Poisson's ratio is taken as .3, these equations may be combined to give

$$S_c = \sqrt{.35\,\frac{f(1/R_1 + 1/R_2)}{L(1/E_1 + 1/E_2)}}$$

where f = force normal to surfaces

L = length of band of contact

If the contact band is taken at the pitch line gear geometry gives radius of contact surface in terms of gear ratio and

$$S_c = \sqrt{\frac{.70}{\left(\frac{1}{E_1} + \frac{1}{E_2}\right) \cos \phi \sin \phi}} \sqrt{\frac{w}{Fd} \quad \frac{m_G+1}{m_G}}$$

where

w = tangential driving load

F = active face width

d = pitch diameter of pinion

$$m_G = \frac{\text{number of teeth in gear}}{\text{number of teeth in pinion}}$$

In the case of steel gears of 20° pressure angle, the left hand term becomes 5715.

As a convenience, many gear engineers use the symbol K for the right hand term.

$$K = \frac{w}{Fd} \left(\frac{m_G+1}{m_G} \right)$$

or

$$S_c = 5714 \sqrt{K}$$

In the case of helical gears, the equation for surface endurance can be written

$$S_c = C_k \sqrt{\frac{K}{m_p}}$$

where

m_p = profile contact ratio

$$C_K = \sqrt{\frac{.70 \cos^2 \phi}{\left(\frac{1}{E_1} + \frac{1}{E_2}\right) \cos \phi_n \sin \phi_n}}$$

Test runs on thousands of gears have proved the validity of these equations on gears operating in conventional environments.

In addition, comparison between theoretical and experimental results of gears of large variations in diameter and tooth size have established that these equations are valid from gears from less than 3 to finer than 24 diametrical pitch and from less than 1 inch to over 100 inches in pitch diameter. Thus, we believed these equations suitable for scaling gears on test rigs to those in specific applications within these size ranges.

It was therefore proposed to test gears of 12 diametrical pitch and of 1-1/4 inch pinion and 4 inch gear in a four square gear tester in the ATL vacuum chamber. These results were scaled by means of these equations to the hoist gears of about equal size and up to the transfer car gears of somewhat larger sizes.

The helical gear test loads were selected from the calculated load ranges indicated in Table 27.

Test Plan

The test plan for the gearing phase of the program was based on the following considerations:

1. A major effort to determine the load carrying capacity of power gearing had been previously conducted by the General Electric Company. One of the principal efforts had been to determine the ability of gear teeth (of all sizes) to withstand the combination sliding stress and Hertzian compressive stress present when a load is transmitted from one gear to its mate. Equations to express the load capacity in terms of gear size and material combinations lubricated by conventional lubricants had evolved, and test data yielding values for the various terms in the equations had been obtained.

   It was proposed to carry out the high vacuum phase of the test program in a similar way so as to extend the already large volume of engineering data into the high vacuum area.

2. The kinematic action of gear teeth meshing together is well understood. This knowledge indicated a possibility of testing gear material-lubricant combinations at a convenient test gear size and extending this data in either direction to apply to gears actually used in a space simulation chamber.

3. A survey of the kinds, and sizes of gears and the loads that will be found in the chamber indicated that test gears of approximately 12 diametrical pitch and of 4 or 6 to 1 ratio are typical of gears in hoist applications. The gears in transfer carts are of somewhat larger size.

The most favorable lubricants were run in a Four-Square Gear Tester located inside of a vacuum chamber. The gears used were of the same size (diametrical pitch and pitch diameter) as some of the smaller gears used in actual simulator equipment. These tests established loads and life cycles that could be achieved by the various lubricant coatings examined.

TABLE 27

ATL GEAR TEST LOADS

Torque on Shaft (in.lbs.)	Tangential Tooth Load (lbs.)	"K" Factor	Surface Compressive Stress
10	16	16.8	16,900
20	32	33.6	24,000
40	64	67.2	33,900
80	128	134.4	47,900
160	256	269	67,700
320	512	538	95,800
640	1,024	1,075	136,000
1280	2,048	2,150	192,000
2560	4,096	4,301	270,000

Note: Based on test gears (ATL)

No. teeth pinion - 15 gear - 48

Transverse pitch - 12

Helix angle - 22-1/2° pressure angle - 20°

SECTION VIII - SCREENING AND 30 MM BEARING TEST RESULTS

A. SCREENING TESTS

Screening tests consisting of materials engaged in sliding contact were conducted on the various candidate lubricants, including those experimental coatings. Screening tests were conducted on the various candidate lubricants for the following reasons:

1. Relative sliding characteristics of thin coatings could be ascertained with respect to:

    a) Coefficient of friction
    b) Wear characteristics
    c) Surface preparation effect, i.e., vapor blast, etc.
    d) Environment, i.e., air vs. dry inert argon

2. Coating processes and techniques could be tried out on relatively simple surfaces, thereby partially establishing feasibility, and necessary processing controls for the more complex bearing and gear surfaces.

3. A number of coatings and test conditions could be examined over a reasonable period of time as well as economically, thereby allowing for optimum use of the bearing and gear test facilities at the Advanced Technology Laboratories and the Arnold Engineering Development Center.

The tests conducted in dry inert argon gas partially simulated the environment that the materials and coatings would normally be subjected to in a vacuum chamber, with the exception of pressure. Further, the majority of candidate coatings examined possessed low vapor pressure characteristics. Evaporation effects, therefore, were not expected to play a significant role in coating performance at this time.

<u>Apparatus and Procedure</u>

The tests were conducted on a wear test apparatus in which a 3/4" long x 3/8" diameter slider, with a hemispherical end consisting of a 3/16" radius test surface, was mounted in a reciprocating arm, and moved in relation to the surface of a stationary, 2" long x 1" wide x 1/4" thick, flat specimen, at a velocity of 3 feet per minute (18 cycles per minute). One end of the arm was so mounted that the free end was able to move in either a vertical or horizontal plane. Sliding was obtained by moving the arm back and forth by means of an air-actuated piston. To measure friction a flexure arm mounted with strain gages was placed between the air piston and the friction arm. This arrangement then gave a continuous recording of the friction force. Measurements, consisting of change in height of lever arm and diameter of wear scar generated on slider hemisphere, were made for each run. The flat specimen, which was held stationary on a supporting table, was housed within an enclosure. Another enclosure, positioned around the slider specimen, was fastened to the moving arm. This allowed for better control of gas blanketing. Clean, dry argon gas was introduced under positive pressure into the enclosed test area to prevent the influx of oxygen and moisture. Some of the tests were conducted in a dry argon environment and some in normal atmosphere, both conditions at room temperature. A counter connected to the air-actuated piston arm recorded the number of cycles run.

Figure 9 is a schematic diagram of the front view of the test apparatus. Figure 10 is a photograph of the test apparatus with the moving arm swung back 180° from its normal operating position in order to show the specimens in their respective housings.

Discussion of Results

The results of the screening tests, documented in Table 28, are summarized briefly as follows:

1. The majority of solid film lubricants investigated exhibited higher wear characteristics in air than in the dry inert argon.

2. Macroscopic and microscopic examinations of the lubricants, in addition to the analysis of the data, indicated promise for such solid films as:

    a. MoS_2 + graphite + silicate binder
    b. MoS_2 + glass binder (ATL development)
    c. 24 kt. gold
    d. Silver
    e. CBS coating
    f. Graphite + $AlPO_4$

3. Pre-treatment of the surfaces by vapor blasting improved the overall performance of the lubricant in most cases.

4. Solid film coatings containing MoS_2 and/or graphite with their associated binders "ran in" easier and performed with less frictional resistance than the thin film platings.

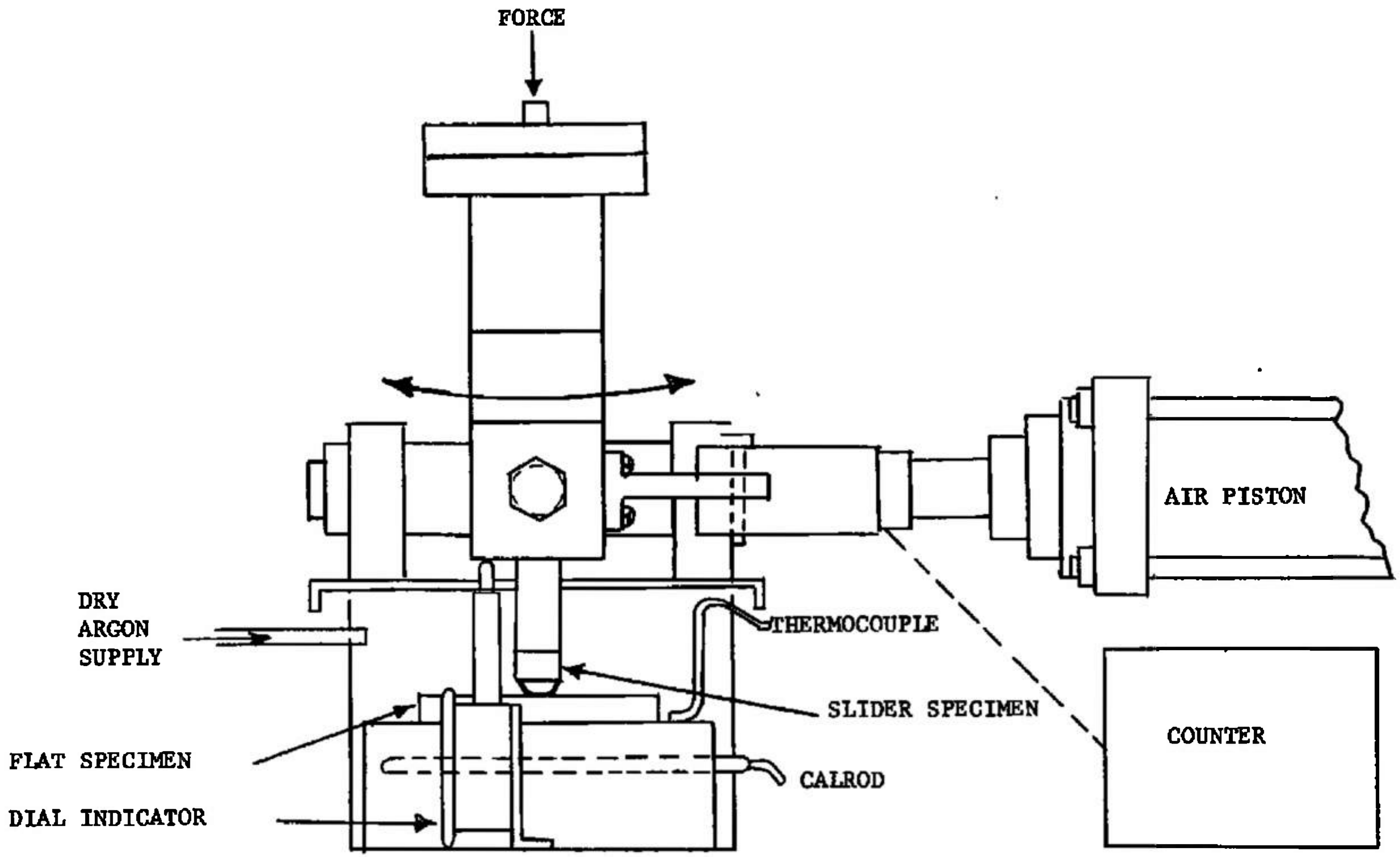

Fig. 9 Schematic Diagram – Front View of Wear Test Apparatus

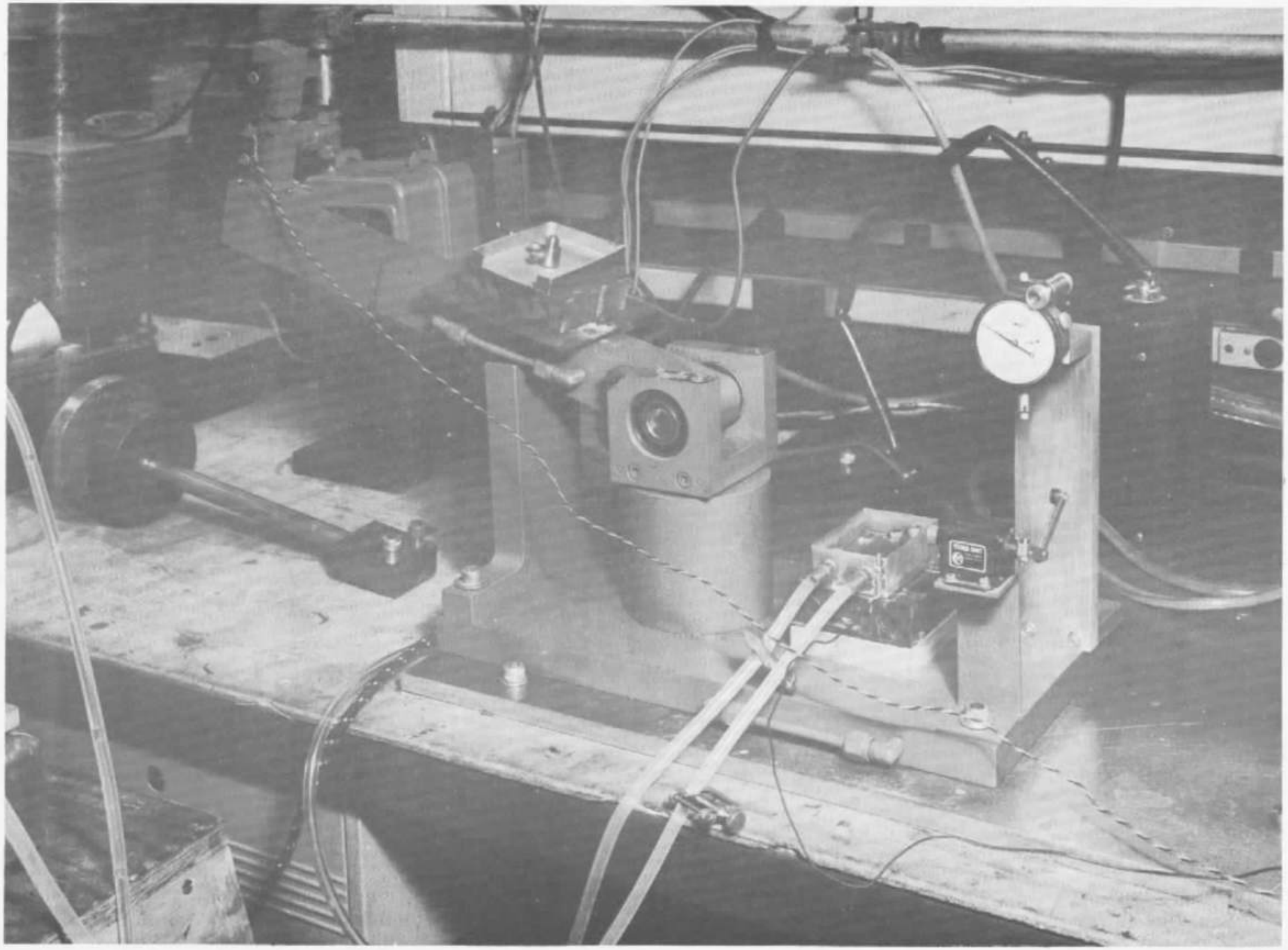

Fig. 10 Reciprocating Wear Test Apparatus, Showing Test Specimens in Their Respective Housings

TABLE 28

SUMMARY OF SCREENING TEST RESULTS

TEST NO.	TEST SPECIMEN(1) LUBRICANT — SLIDER	TEST SPECIMEN(1) LUBRICANT — FLAT	FLAT SURFACE PRE-TREATMENT(2)	SLIDER WEAR SCAR DIA. (Mils) DRY ARGON A(3)	SLIDER WEAR SCAR DIA. (Mils) AIR B(4)	WEAR INCREASE IN AIR OVER DRY ARGON (%)	COEFFICIENT OF FRICTION AT 100 CYCLES (f) DRY ARGON A(3)	COEFFICIENT OF FRICTION AT 100 CYCLES (f) AIR B(4)
1	23 kt. Gold	23 kt. Gold (Vendor X)	None	50	70	40	0.48	0.48
2	23 kt. Gold	23 kt. Gold (Vendor X)	vapor blasted	54	67	24	0.48	0.40
3	23 kt. Gold	23 kt. Gold (Vendor Z)	vapor blasted	46	68	48	0.44	0.38
4	23 kt. Gold	23 kt. Gold (Vendor Z)	vapor blasted	50	67	34	0.44	0.41
5	none	60% Graphite+40% $AlPO_4$	passivated	32	52	63	0.02	0.03
6	none	60% Graphite+40% $AlPO_4$	none	34	49	44	0.21	0.14
7	none	80% Graphite+20% $AlPO_4$	passivated	32	50	56	0.01	0.16
8	none	80% Graphite+20% $AlPO_4$	sanded	32	47	47	0.68	0.24
9	CBS	CBS	none	15	25	67	0.05	0.05
10	none	none	none	59	71	17	0.70	0.70
11	none	MoS_2+Graphite+Silicate	none	24	13	--	0.05	0.10
12	none	MoS_2+Graphite+Silicate	sand blasted	18	*	--	0.05	*
13	none	CBS	none	13	59	354	0.03	0.07
14	Silver	Silver	vapor blasted	56	66	18	0.20	0.34
15	24 kt.Gold	24 kt. Gold	vapor blasted	42	69	64	0.38	0.52
16	24 kt.Gold	Silver	vapor blasted	50	69	38	0.22	0.38
17	Tin	Tin	vapor blasted	55	71	29	0.68	0.74
18	none	MoS_2+Glass	vapor blasted	49	59	20	0.22	0.20
19	none	MoS_2+Glass (high S)	vapor blasted	39	58	49	0.26	0.14
20	none	MoS_2+Glass	vapor blasted	--	33	--	--	0.05
21	none	MoS_2+Glass (high S)	vapor blasted	--	11	--	--	0.03
22	none	60% Graphite+40% $AlPO_4$	none	--	12	--	--	0.04
23	none	80% Graphite+20% $AlPO_4$	sanded (manual)	--	8	--	--	0.04
24	none	Tin	vapor blasted	--	49	--	--	0.02
25	none	23 kt. Gold	none	--	40	--	--	0.13
26	none	Silver	vapor blasted	--	34	--	--	0.10
27	none	24 kt. Gold	vapor blasted	--	48	--	--	0.10
28	none	Bismuth	none	--	52	--	--	0.10
29	none	Bismuth + Silver	none	--	50	--	--	0.14

FOOTNOTES

(1) Test specimens consisted of AISI 52100 steel hardened to Rockwell C 60, 8-16 rms finish.

(2) Prior to coating specimens were washed in acetone, cleaned with wet levigated alumina on a soft cloth; rinsed in alcohol and distilled water.

(3) A designates tests conducted in dry argon, i.e., 1A, 2A, etc.
B designates tests conducted in air, i.e., 1B, 2B, etc.

(4) Test Conditions:

Tests 1 to 19 -- 15# load, 2000 cycle test duration
Tests 20 to 29 -- 5# load, 1000 cycle test duration

(5) Coating thicknesses (general):

Platings	0.0001-0.0002 in.
CBS coating	4000 angstroms
MoS_2 and Graphite types	0.00015-0.0002 in.

* No Test

B. 30 MM BEARING TESTS

The small scale rolling element bearing tests reported in this section consist essentially of testing spherical and cylindrical type roller bearing. Each test was comprised of a three unit bearing system, that is, a combination of two cylindrical bearings and one spherical, or three of the same kind. One test was run with a PTFE-glass fibre four-sleeve bearing system for a short period of time.

Bearing Materials

The type, and material make-up of the 30 mm bore test bearings can be seen in Table 29. These were stock bearings that were readily available from the bearing vendors.

Test Equipment

The apparatus consists of a vertical drive shaft mounted through support bearings inside a vacuum chamber. A "U" type housing contains the outer races, and load screw used to apply the radial load. Figure 11 is a schematic of the Advanced Technology Laboratories small scale test apparatus, in this case showing a four-test bearing arrangement. A three bearing arrangement was used in these tests, however, in which one bearing (rather than two)took the major load from the load screw. Each end bearing, therefore, was the recipient of one-half the radial load applied to the center bearing. When two cylindrical and one spherical bearing constituted the test bearing system, the spherical bearing was used as the center bearing, therefore, receiving the major load.

The torque of the bearing assembly was obtained by restraining the "U" housing with a cantilever flexure on which strain gages were mounted.

Torque was recorded on a photo-electric recorder.

Radial load was applied by a load screw with calibrated strain gages mounted to measure tension.

Temperature was monitored on the outer race of the loader bearing and bearing housing and recorded on a Leeds and Northrup Speedomax recorder.

The test shaft was driven by a magnetic coupling on either side of the chamber top plate.

Speed control was accomplished by means of a thymotrol motor with speeds up to 30 rpm accomplished through a gear box with a 20:1 ratio.

Results

A summary of the results of the bearing tests can be seen in Table 30. The loads indicated in Table 30 were the maximum radial loads applied to the center bearing, the two end bearings receiving half this amount. Figure 12 shows the torque curves for unlubricated bearings operating in vacuum and air.

TABLE 29

TEST BEARING MATERIALS (30mm)

Type	Specification	Rollers/Balls	Races	Retainers
Cylindrical Roller	E-1206-B Rollway	Case Hardened AISI 8620	52100	SAE 1010
Spherical Roller	22206HL Norma Hoffman	AISI 52100	52100	Brass-58Cu 1.3 Pb, Bal Zn
Sleeve	Fabroid	Four PTFE-glass fibre bearings.		

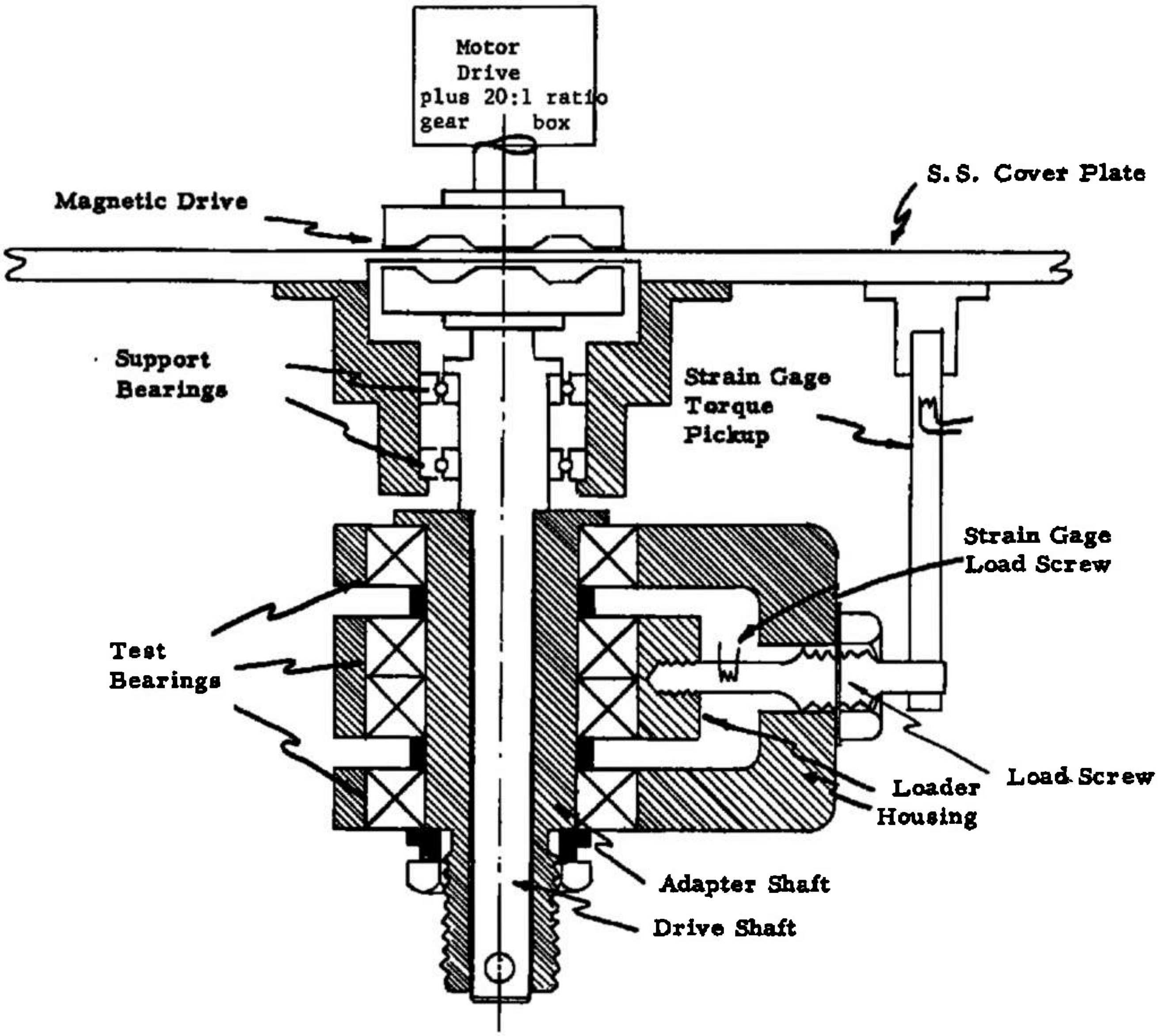

Fig. 11 Schematic: Test Rig

TABLE 30

BEARING TEST RESULTS (30mm)

TEST NO.(1)	BEARING LUBRICANT	BEARING ELEMENT COATED(2)	MAX. RADIAL LOAD ON LOAD BEARING (lbs.)	TEST ENVIRONMENT(3)	TOTAL CYCLES RUN
1	none	none	3400	Air	500
2	none	none	3400	Vacuum	2500
3	24 kt gold	Races, rollers and retainer	3400	Vacuum	120
4	Tin	Races, rollers and retainer	3400	Air	*
5	24 kt gold	Races, rollers and retainer	500	Air	3376
6	MoS_2+graphite + silicate	Races and retainer	2600	Air	51,530
7	Silver	Races, rollers and retainer	1500	Air	6600
8	Silver Gold	Races, rollers and retainer	1000	Air	4050
9	MoS_2+graphite + silicate	Races and retainer	2500	Air/Vacuum	36,750
10	PTFE + glass fibres (sleeve bearings)	Sleeve Bearing	100	Air	5700
11	MoS_2 + graphite + silicate	Races and retainer	2000	Air	48,050
12	MoS_2 + graphite + silicate	Races and retainer	1000	Air	15,000
13	24 kt gold	Races, rollers and retainer	500	Air	9300
14	Graphite + $AlPO_4$	Races and retainer	1000	Vacuum	16,650
15	MoS_2 + glass	Races and retainer	1200	Air/Vacuum	41,750

* Jammed on start.

Footnotes follow

FOOTNOTES FOR TABLE 30 - BEARING TEST RESULTS (30mm)

(1) All bearing systems tested consisted of a combination of two cylindrical and one spherical bearings with the following exceptions:

Test 5 - 3 spherical bearings
Test 10 - 4 PTFE + glass fibre sleeve bearings
Test 12 - 3 cylindrical bearings
Test 13 - 3 spherical bearings

(2) With exception of Test Nos. 1, 2, and 10, races and retainer surfaces were vapor blasted prior to coating.

(3) Tests in vacuum were conducted from 10^{-4} to 10^{-6} mm Hg pressure.

(4) All tests were run at 30 rpm with the exception of Tests 1-5, run at 20 rpm.

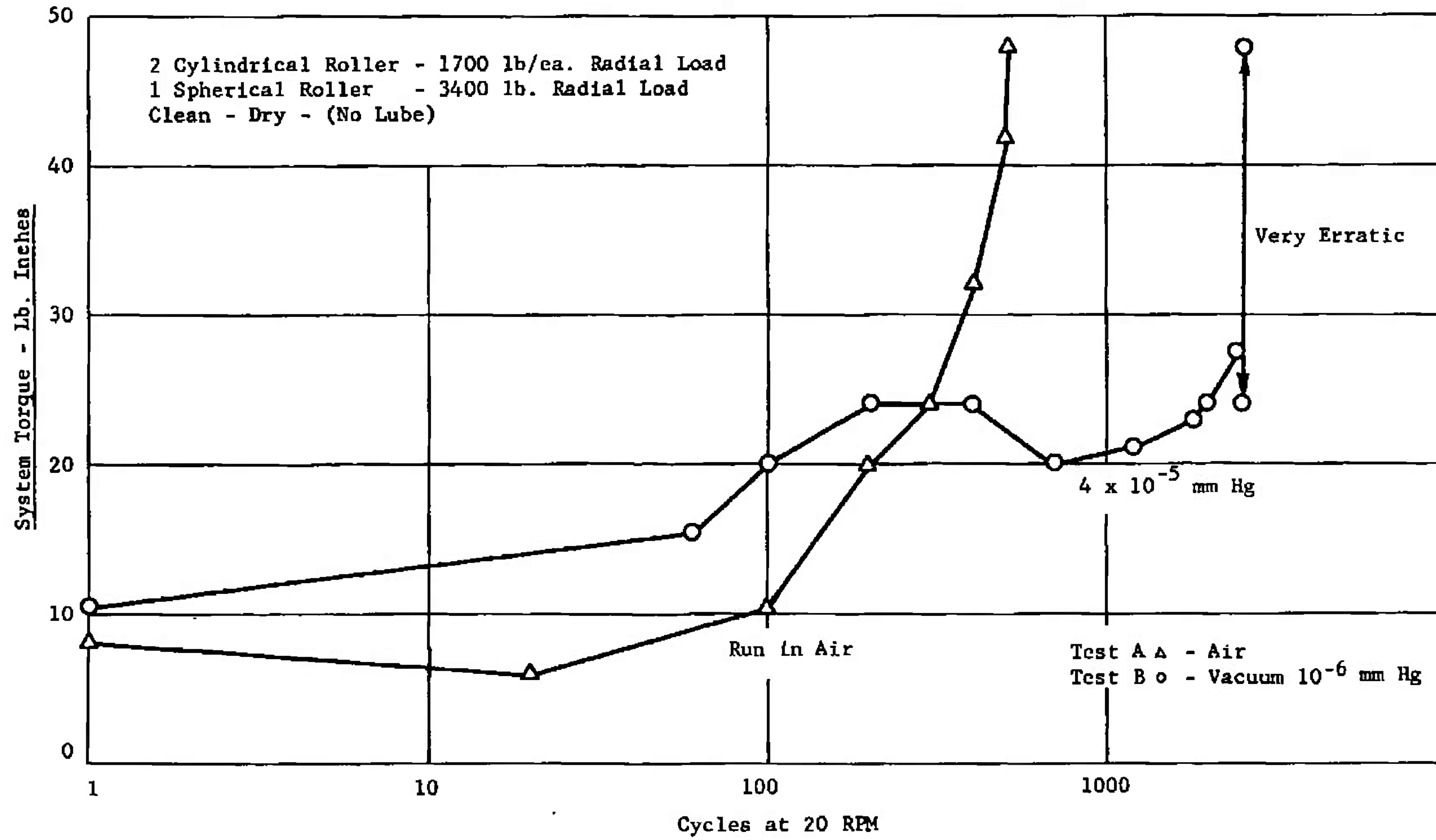

Fig. 12 Comparison of 30mm Bearing Performance in Vacuum and Air Environment

Discussion of Results

The longest run time periods in air were obtained with the solid films of MoS_2- Graphite-Silicate, experimental MoS_2-glass, and experimental Graphite-aluminum phosphate. The silver and gold metallic films followed. Tin plate performed poorly. The PTFE + glass sleeve bearings were extremely sensitive to load and at 50 lb/brg exhibited high torque.

The criteria established for suspending a test was a rapid increase in system torque generally to the point approaching magnetic drive coupling slippage (approximately 42 lb-in) or outright bearing jamming.

As a further result of this phase, bearings for future testing were increased in size (see 50 mm bearing test results in Section IX) and the ratio of bearing static capacity (C_o) to applied radial load (P) was increased from three to the order of ten.

SECTION IX – SMALL SCALE BEARING (50MM) AND GEAR TEST RESULTS

INTRODUCTION

A series of 50 mm cylindrical, spherical and tapered rolling element bearing tests were conducted using as a basis selective lubricants that had previously been examined in sliding tests reported in Section VIII.

The lubricants applied to the test bearings consisted of:

1. 23 kt gold
2. 24 kt gold
3. Silver
4. MoS_2 + epoxy binder
5. MoS_2 + graphite + sodium silicate binder
6. MoS_2 + glass binder (dev)*

Low vapor pressure greases:

7. Petroleum distillate-Apiezon L
8. Methyl chlorophenyl silicone + 5% MoS_2 (dev)*
9. Polyphenyl ether (dev)*

The first part of this section is devoted to these small scale tests which were the forerunners of the larger bearing tests which the Advanced Technology Laboratories conducted in the AEDC aerospace chamber.

The second part of Section IX describes the small scale gear tests and their respective results. Actually, the gear test size selected met the size of many of the gears that will be used in the actual simulator application.

The lubricants applied to the test gears consisted of:

1. 23 kt gold
2. Silver
3. MoS_2 + epoxy binder
4. MoS_2 + graphite and silicate binder

The test apparatus, vacuum chamber, and test results for the small scale bearing and gear tests are all described in this section.

A. SMALL SCALE BEARING TEST APPARATUS

The bearing test apparatus consisted essentially of a vertical drive shaft mounted through support bearings inside a vacuum chamber. A yoke or "U" type housing contained the outer races while the innermost test bearings were contained in a housing adapted to accommodate a load screw by which radial loads could be applied. The apparatus was adaptable to accommodate three or four test bearings at one time.

*Advanced Technology Laboratories developmental effort

A relatively clear conception of the bearing test apparatus can be seen in the Figure13 schematic, and Figure 14 photograph. The "system" torque of the bearing assembly was obtained by restraining the "U" housing with a cantilever flexure on which strain gages were mounted. Torque was recorded on a photo-electric recorder.

Radial load was applied by a load screw on which strain gages were mounted to record tension. The load bolt was calibrated in pounds tension vs. indicator micro-strain. Temperature was monitored on the outer race of the loader bearings and bearing housing and recorded on a Leeds and Northrup Speedomax recorder.

Heaters mounted in the bearing housing maintained a constant 75° F environmental test temperature. The test shaft was driven by a magnetic coupling on either side of the chamber top plate. The desired test speeds were obtained by the use of a thymotrol speed control motor operating in series with a 20:1 reduction gear box.

Vacuum Chamber

The vacuum system consisted of a stainless steel chamber 2 feet in diameter by 3 feet, 6 inches in length. The chamber incorporated as an integral part of a liquid nitrogen cold trap with a capacity of approximately 12 liters leaving a net working depth of 2 feet. This trap acted as an anti-migration feature to reduce migration of diffusion pump oil which would ordinarily migrate along the warm walls into the chamber. In addition, to further reduce this tendency, an auxiliary cylindrical-walled liquid nitrogen trap was used which completely surrounded the test apparatus. The pumping system consists of a 6 inch CEC diffusion pump and a 46 CFM Kinney Mechanical pump. The diffusion pump is connected directly to the chamber beneath the cold trap. A water cooled baffle plate was located directly above the inlet to the pump preventing hot oil vapors from diffusing back into the condensing area. Flanged connections on the high vacuum side of the diffusion pump were provided with double "O" rings and pump out lines thus controlling to a great extent leakage which otherwise is undetected. The top of this chamber incorporates this same arrangement.

The ATL vacuum chamber used for the small scale tests can be seen in the Figure 15 photograph, and Figure 16 schematic.

Instrumentation

A schematic of the instrumentation setup can be seen in Figure 17. All leads were brought out through hermetically sealed glass to Kovar feed-throughs.

B. BEARING TEST RESULTS

Interpretation of System Torque Curves

Abnormal variation in system torque during bearing operation gave rise to the possibility of coating or plating, transfer, buildup, or lubricant marginality. The limiting torque, based on the magnetic drive system was

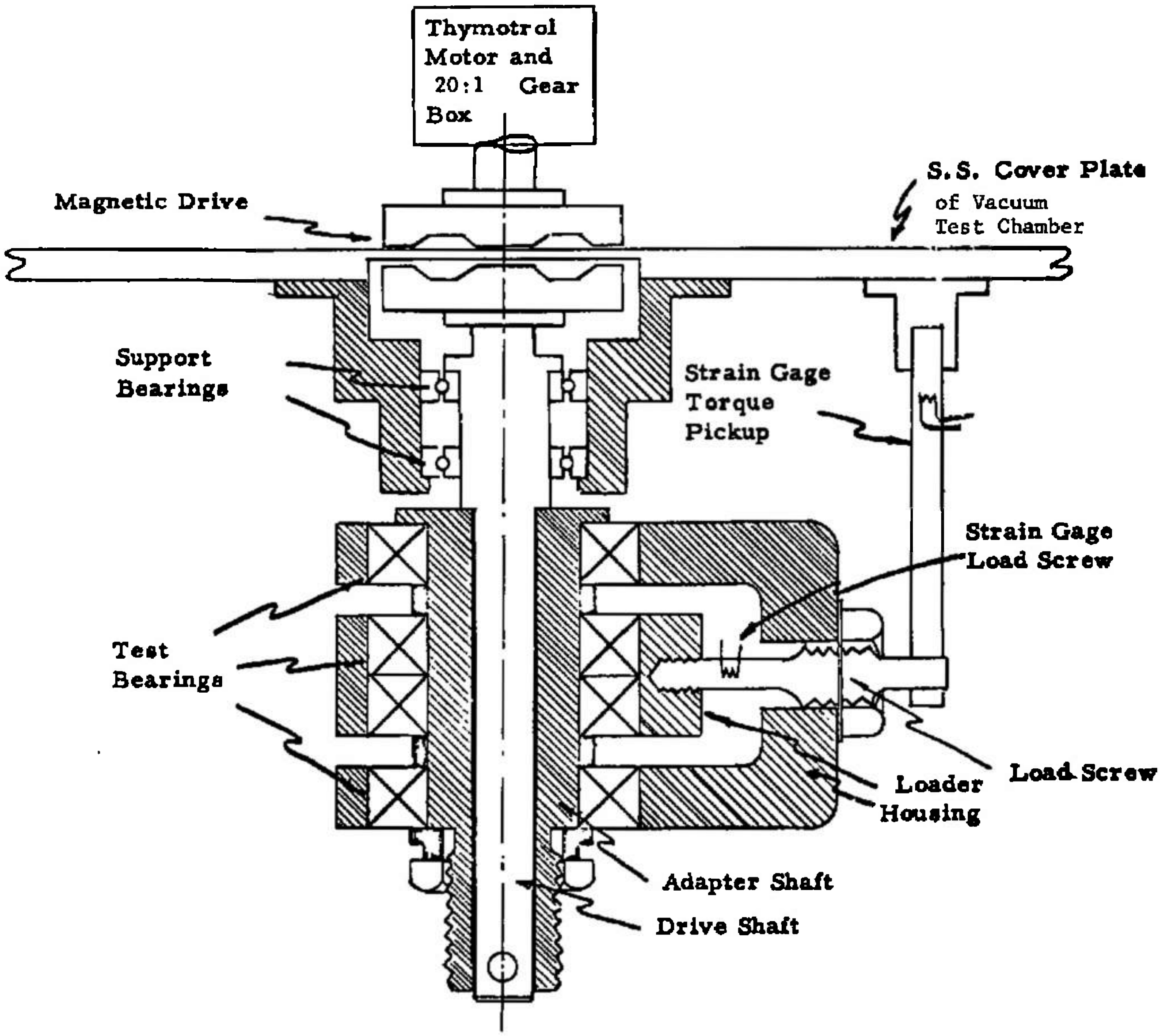

Fig. 13 Schematic: Test Rig

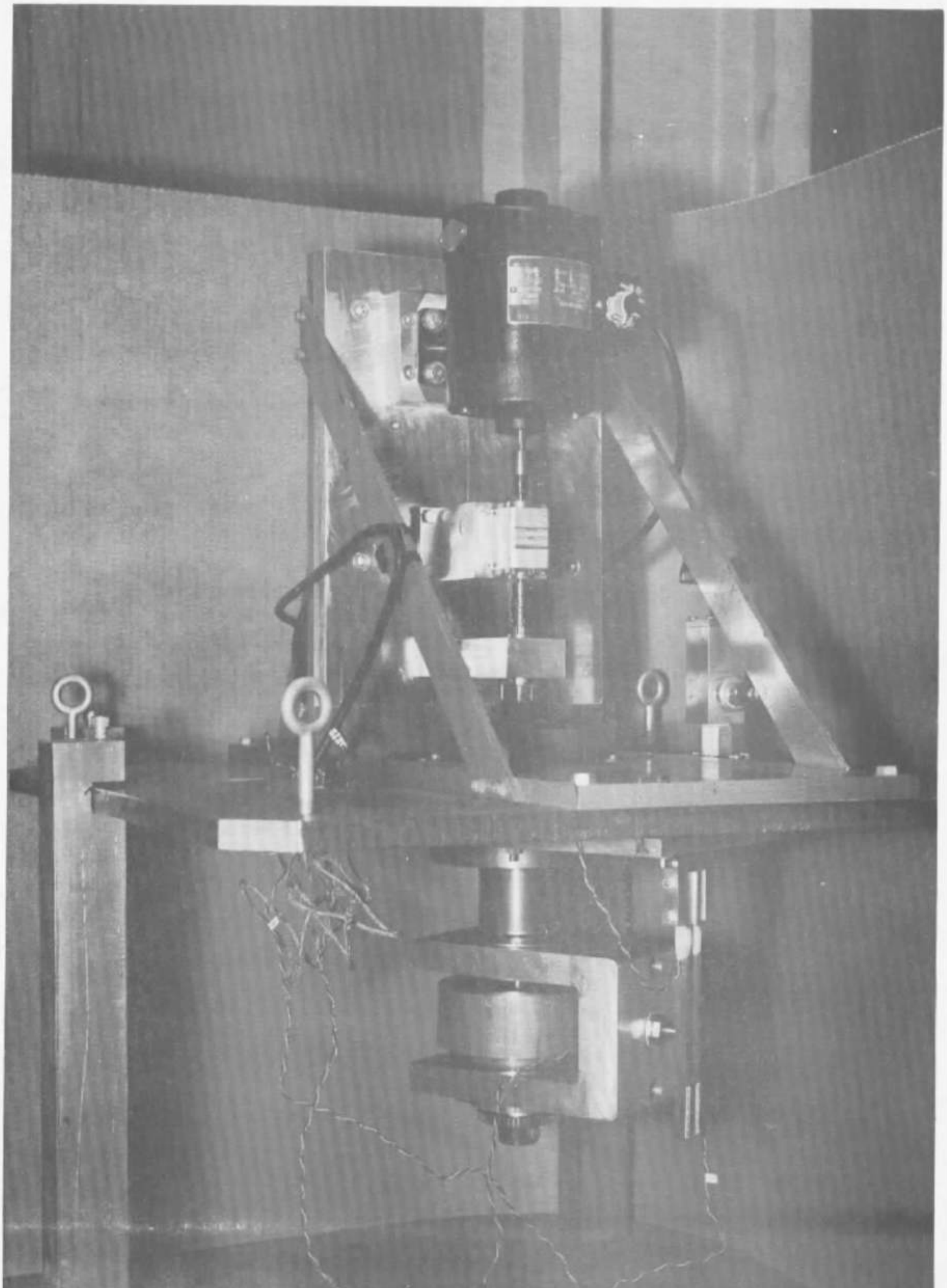

Fig. 14 Small Scale Bearing Test Apparatus

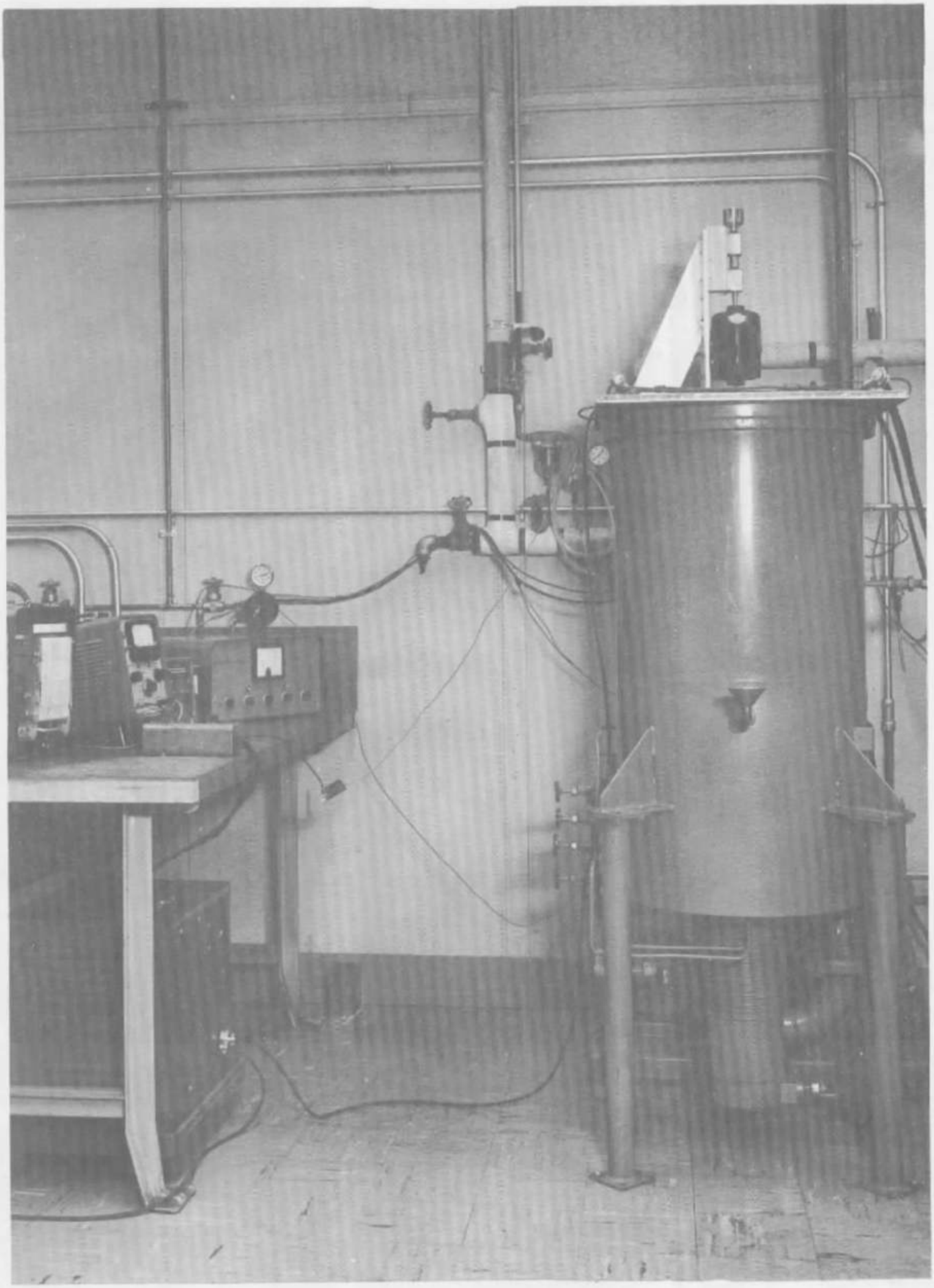

Fig. 15 Vacuum Test Facility for Small Scale Tests

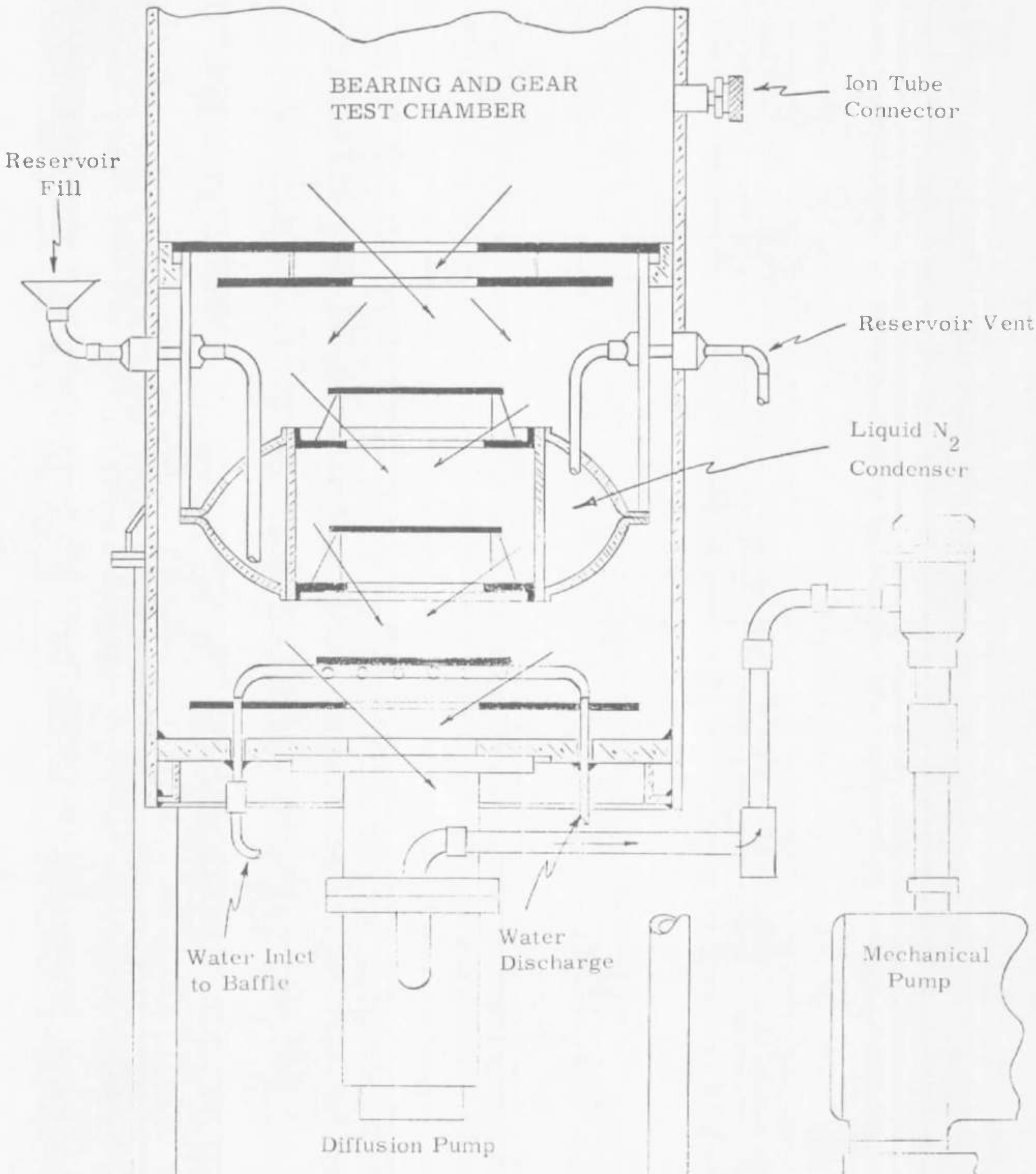

Fig. 16 Schematic of Vacuum System

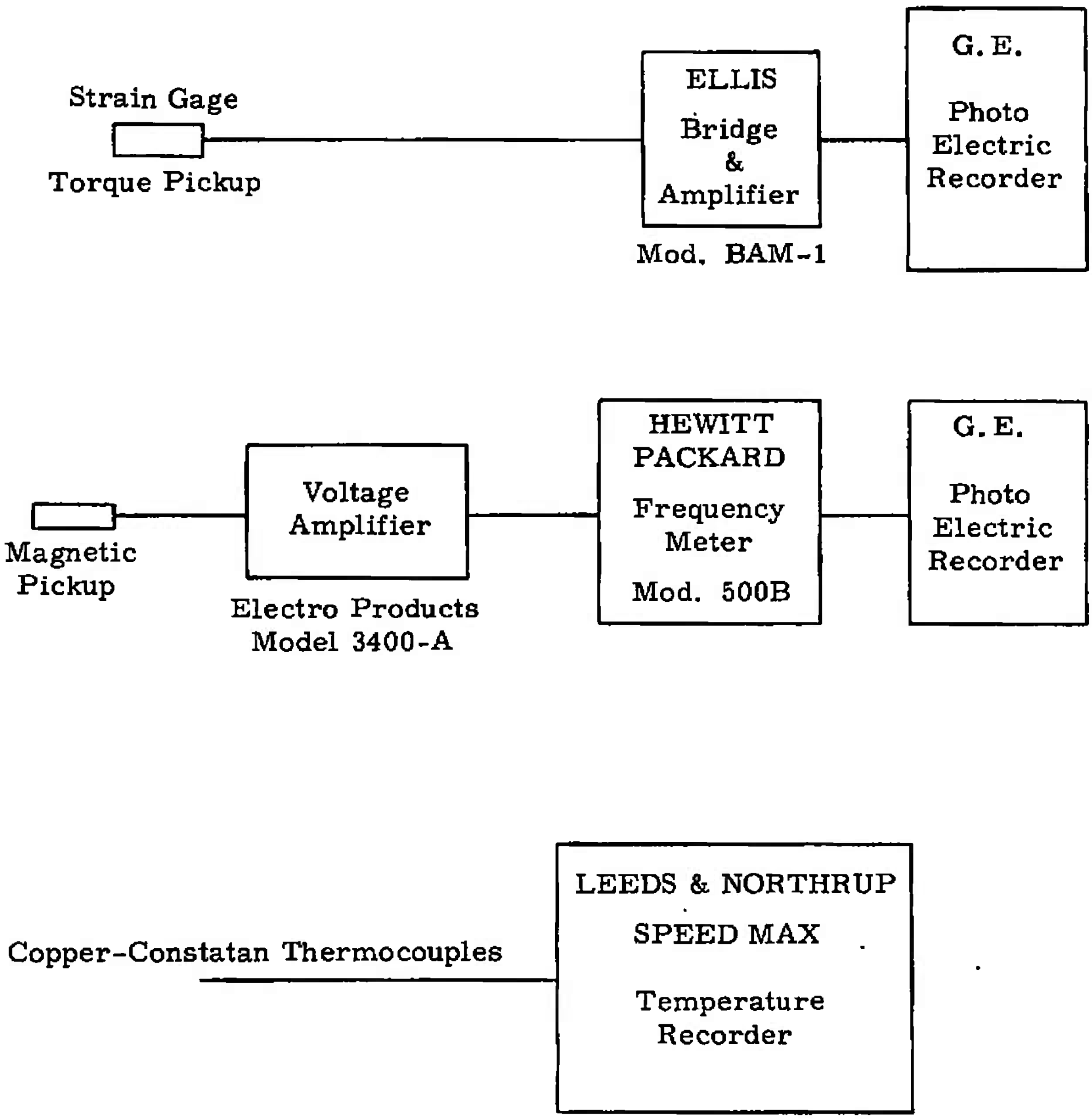

Fig. 17 Schematic of Instrumentation

42 lb-in. The magnetic clutch inside the chamber would slip when this value was exceeded. Hence, the sensitivity of the system torque measurements was quite good. Subsequent macroscopic and microscopic examinations of the tested bearings bore this out as sharp increases in system torque meant a degree of lubricant transfer, buildup and/or wear had occurred rather than indicating a catastrophic type of failure.

Further, approaching the 42 lb-in. torque drive limit did not necessarily indicate that the test bearings could not be operated for an additional period as a small amount of wear debris starting to build up in some cases would be deformed uniformly over the bearing surface thereby allowing the bearing to operate further at a higher torque level, or in some cases dropping the torque close to the initial level. The same mechanism could exist at torque levels in excess of the 42 lb-in. test equipment limit.

Hence, the criterion for terminating a bearing test was based on a sharp increase in torque (approaching magnetic drive slippage), which was interpreted as the first sign or indication of lubricant wear, buildup or transfer, and not necessarily catastrophic failure. This allowed the bearing elements to be examined while the lubricants were intact in varying degrees consequently developing the performance characteristics of the lubricant in question. For the torque curves of the various 50 mm bearing-lubricant systems see Figures 18 - 27 "Average System Torque".

A summary of the bearing test results can be seen in Table 31. 50,000 cycles of operation, and over, was exhibited by such coatings as silver on Timken tapered roller bearings, MoS_2 + graphite + silicate, experimental MoS_2 + glass, on Timken tapered spherical and cylindrical roller bearings. Apiezon L, used for comparison purposes, performed exceedingly well. The quantities of low vapor pressure greases used can be seen in Table 32 and a weight change analysis based on a static 500 hr exposure to vacuum, in Table 33 .

It will be noted in Table 34 , "Examination Results of Small Scale Bearing Tests (50MM)", that some of the bearings had their thin film lubricant intact at the conclusion of the test even though their torque curves showed sharp rises. This condition further attests to the need for "loose" bearing clearances, (see Section V, Table 20). It should be noted that the initial bearing clearance must be adequate to allow for coating the respective bearing elements as well as allowing the bearing to rotate freely upon assembly. Too tight a clearance can cause immediate jamming, particularly with thin film platings. A reasonably loose clearance (after coating) gives the bearing elements a chance, upon initial operation, to remove, relocate, and/or deform small amounts of coating wear debris, resulting in what is referred to as "run-in".

The mode of wear behavior of various types of thin film lubricants are effectively described throughout Table 34 .

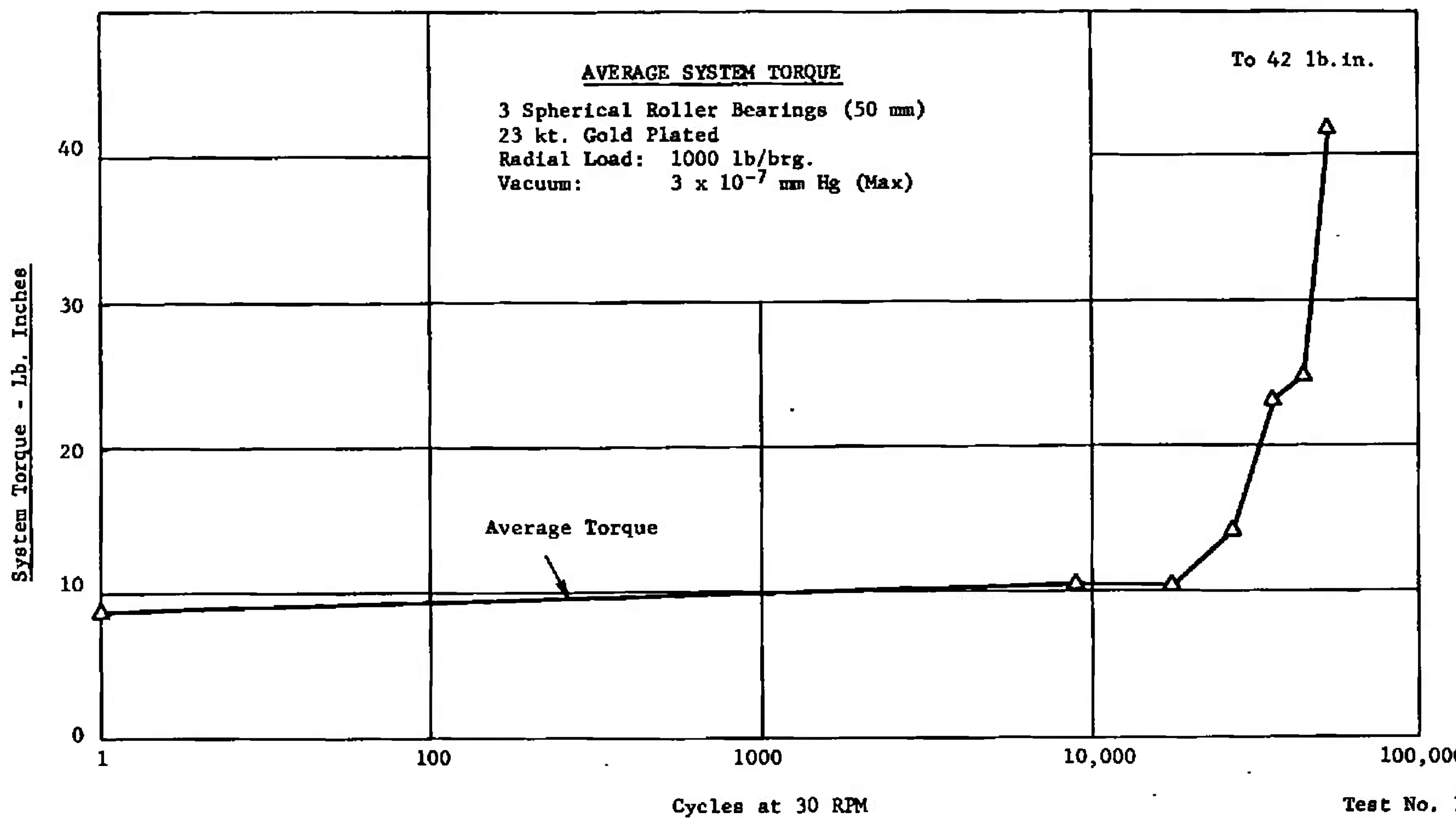

Fig. 18 Average System Torque Curves of the Various 50 mm Bearing-Lubricant Systems

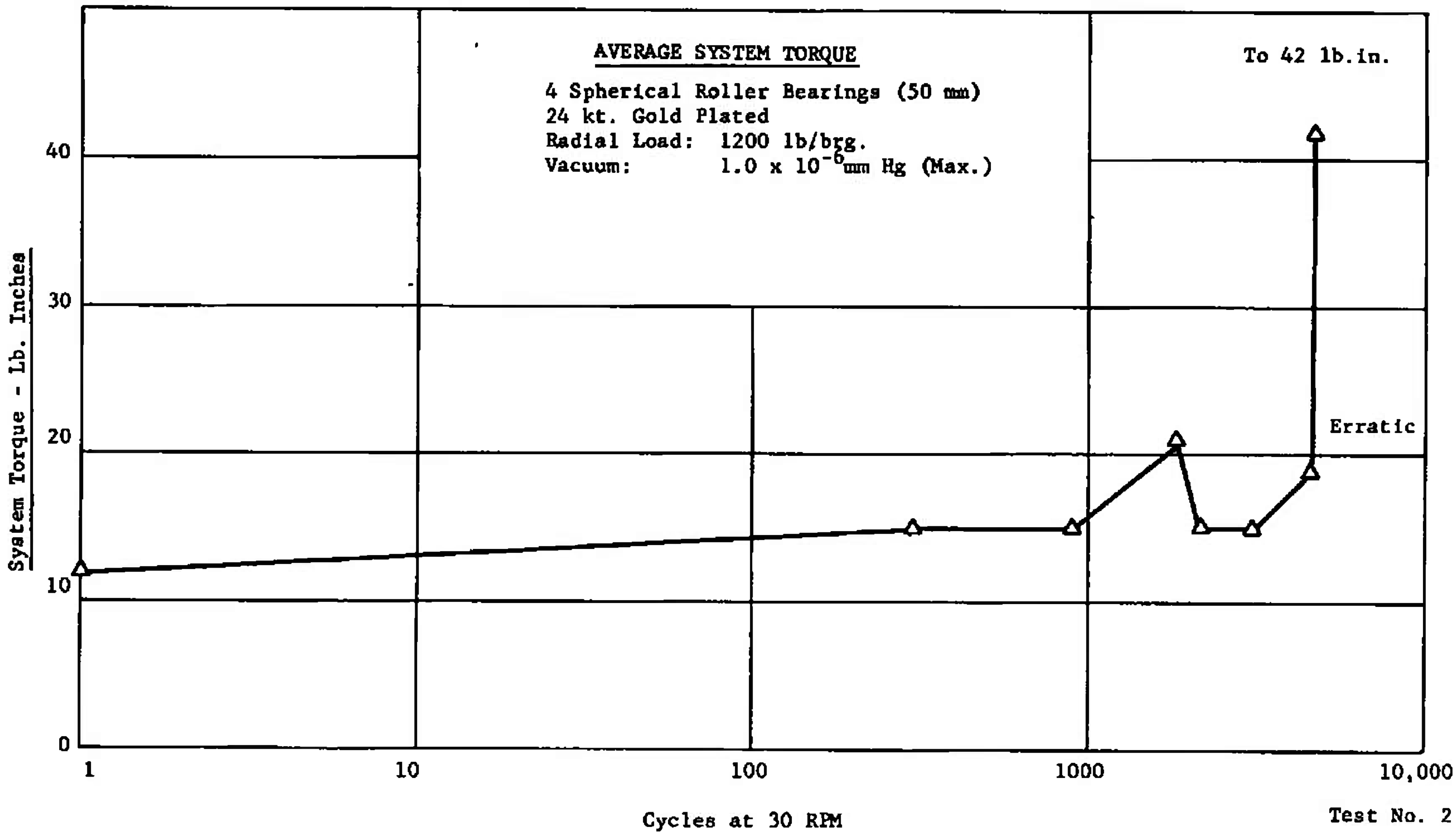

Fig. 19 Average System Torque Curves of the Various 50 mm Bearing-Lubricant Systems

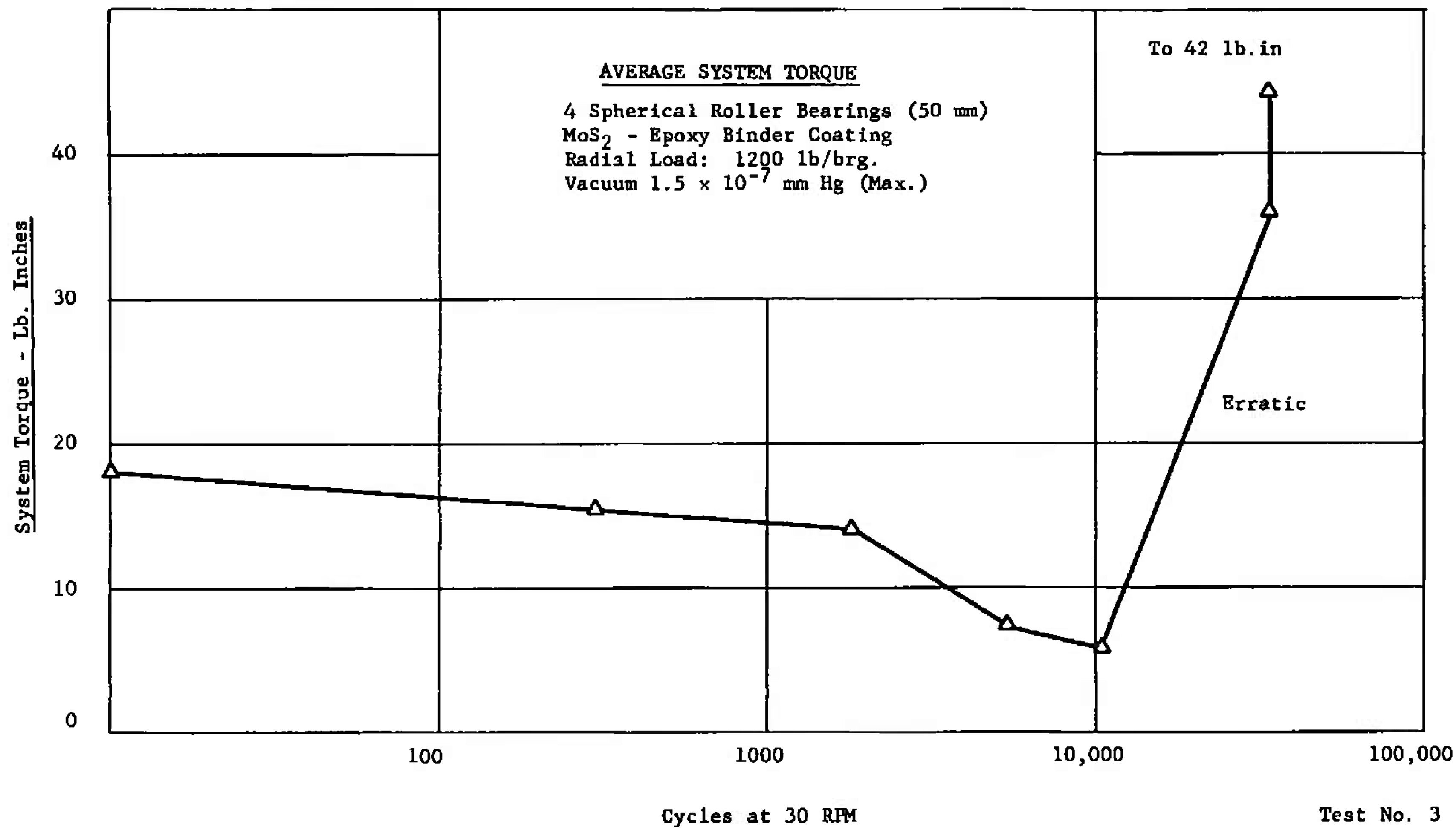

Fig. 20 Average System Torque Curves of the Various 50mm Bearing-Lubricant Systems

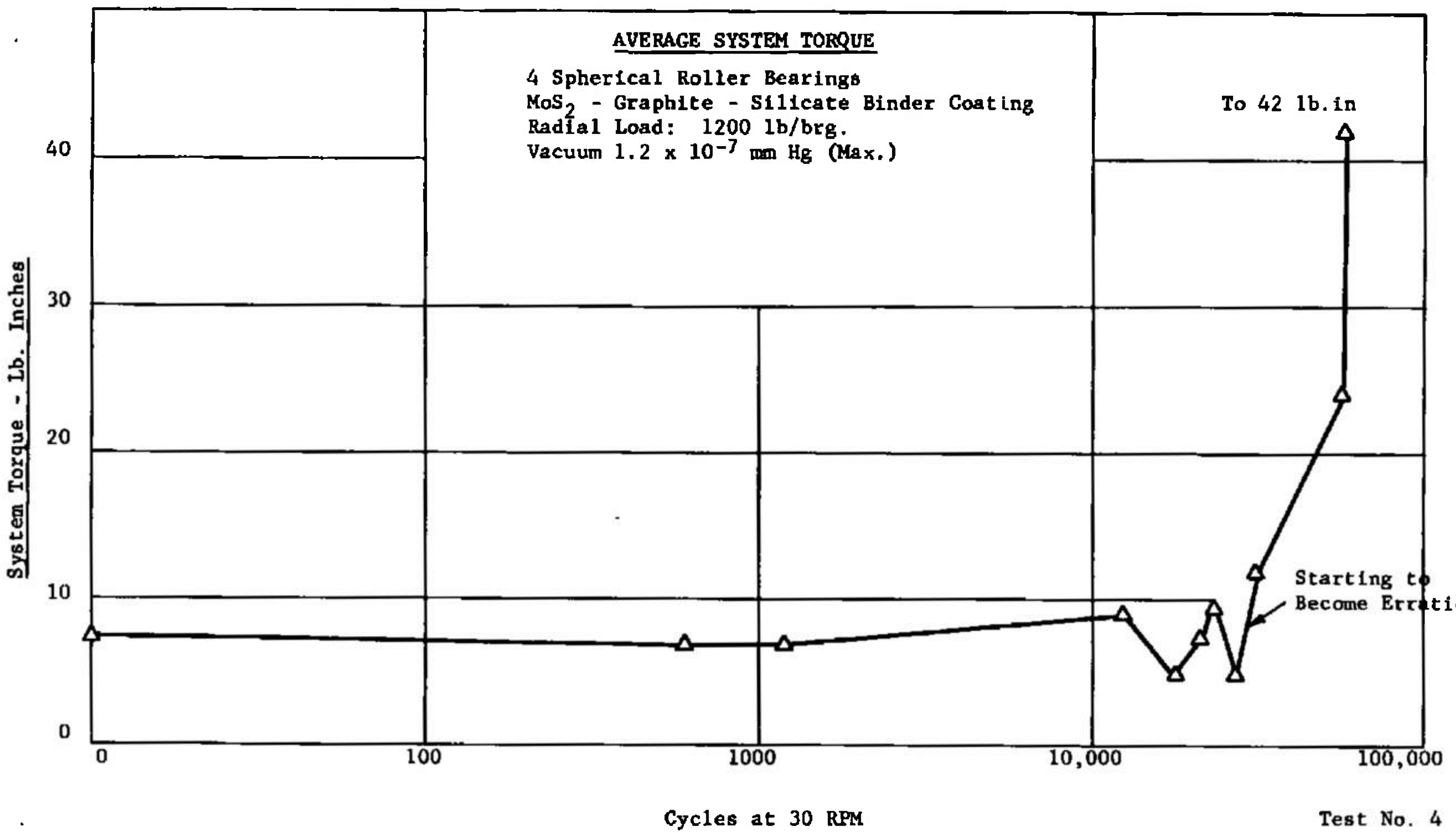

Fig. 21 Average System Torque Curves of the Various 50mm Bearing-Lubricant Systems

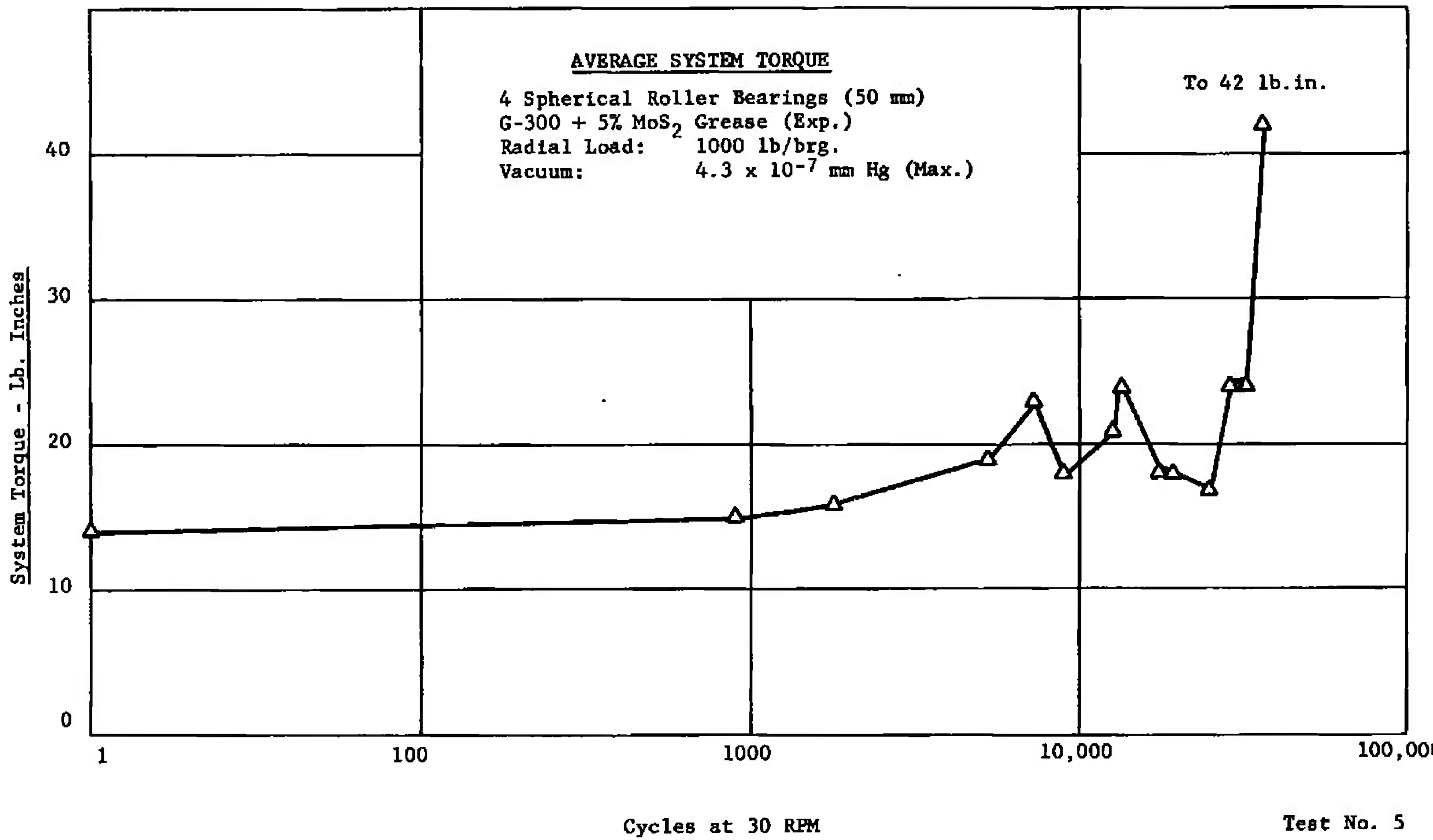

Fig. 22 Average System Torque Curves of the Various 50mm Bearing-Lubricant Systems

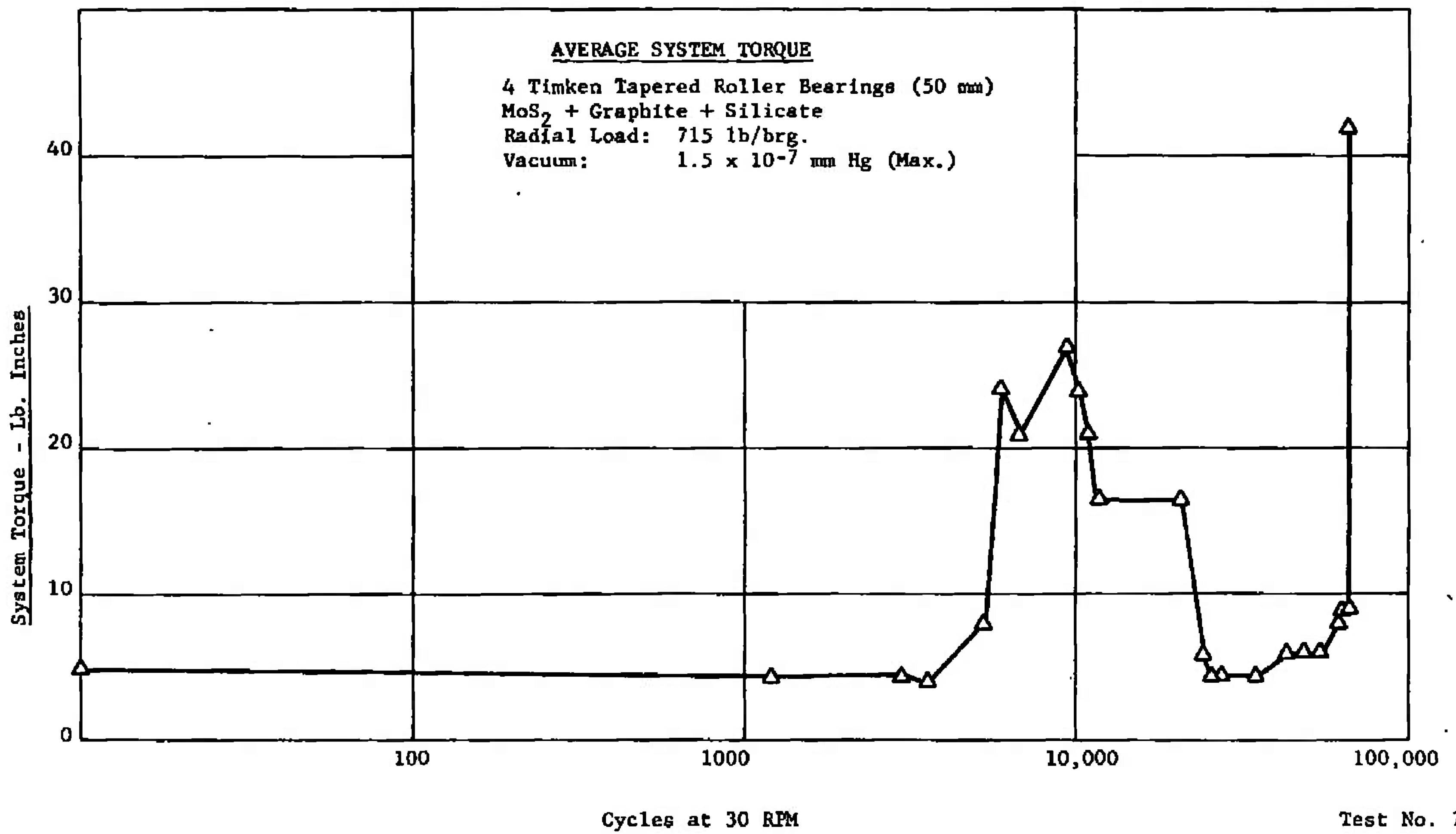

Fig. 23 Average System Torque Curves of the Various 50mm Bearing-Lubricant Systems

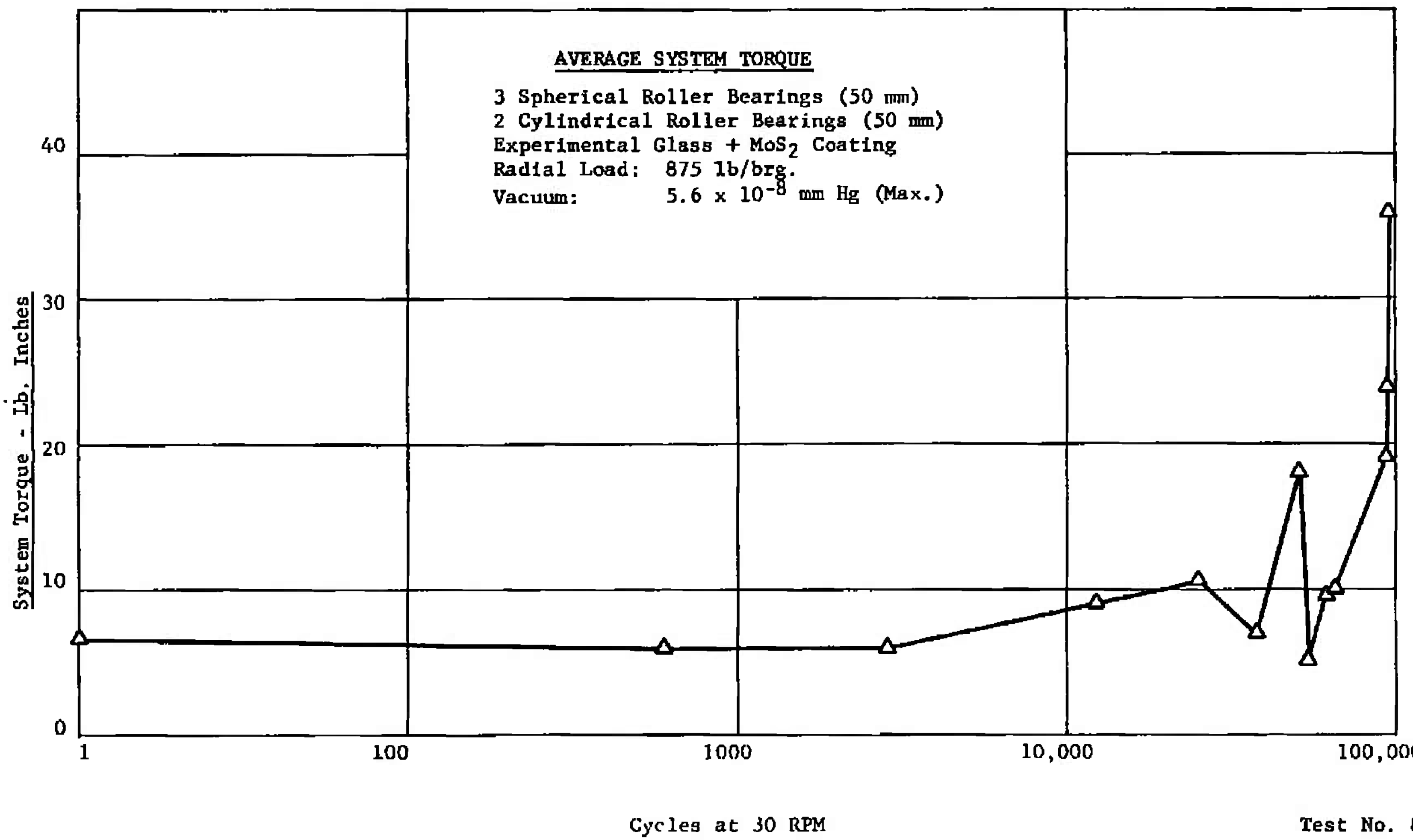

Fig. 24 Average System Torque Curves of the Various 50 mm Bearing-Lubricant Systems

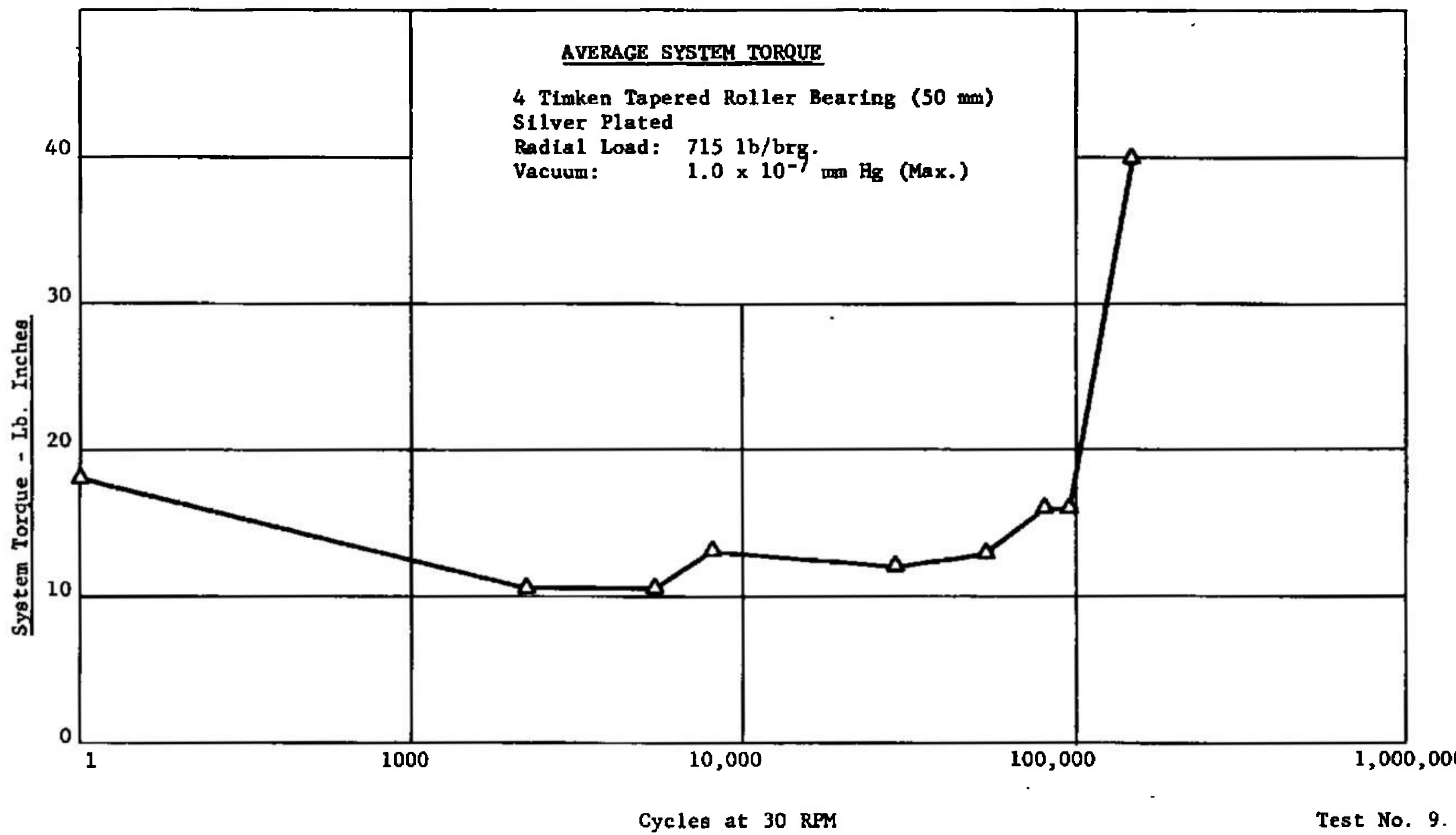

Fig. 25 Average System Torque Curves of the Various 50mm Bearing-Lubricant Systems

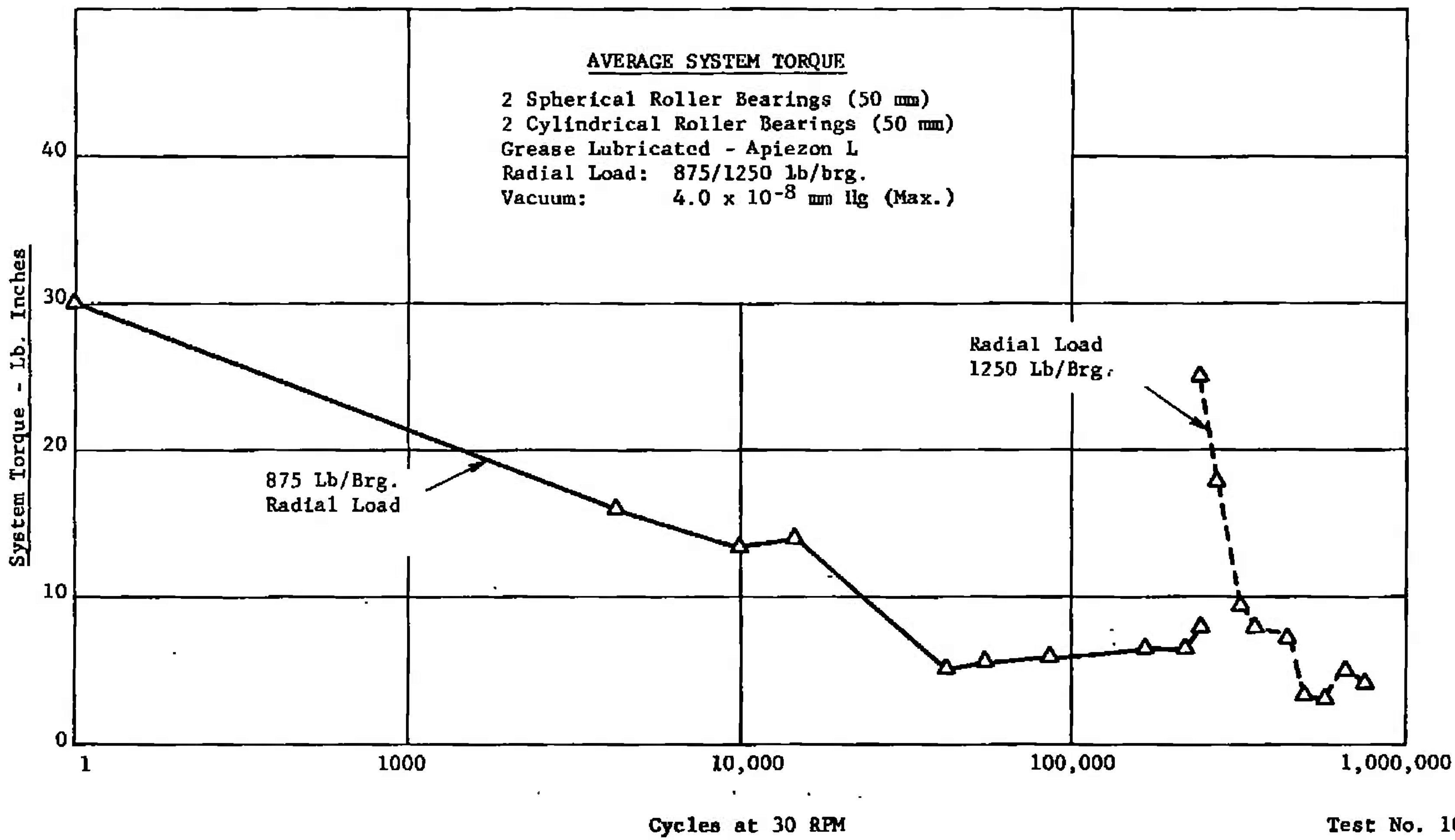

Fig. 26 Average System Torque Curves of the Various 50mm Bearing-Lubricant Systems

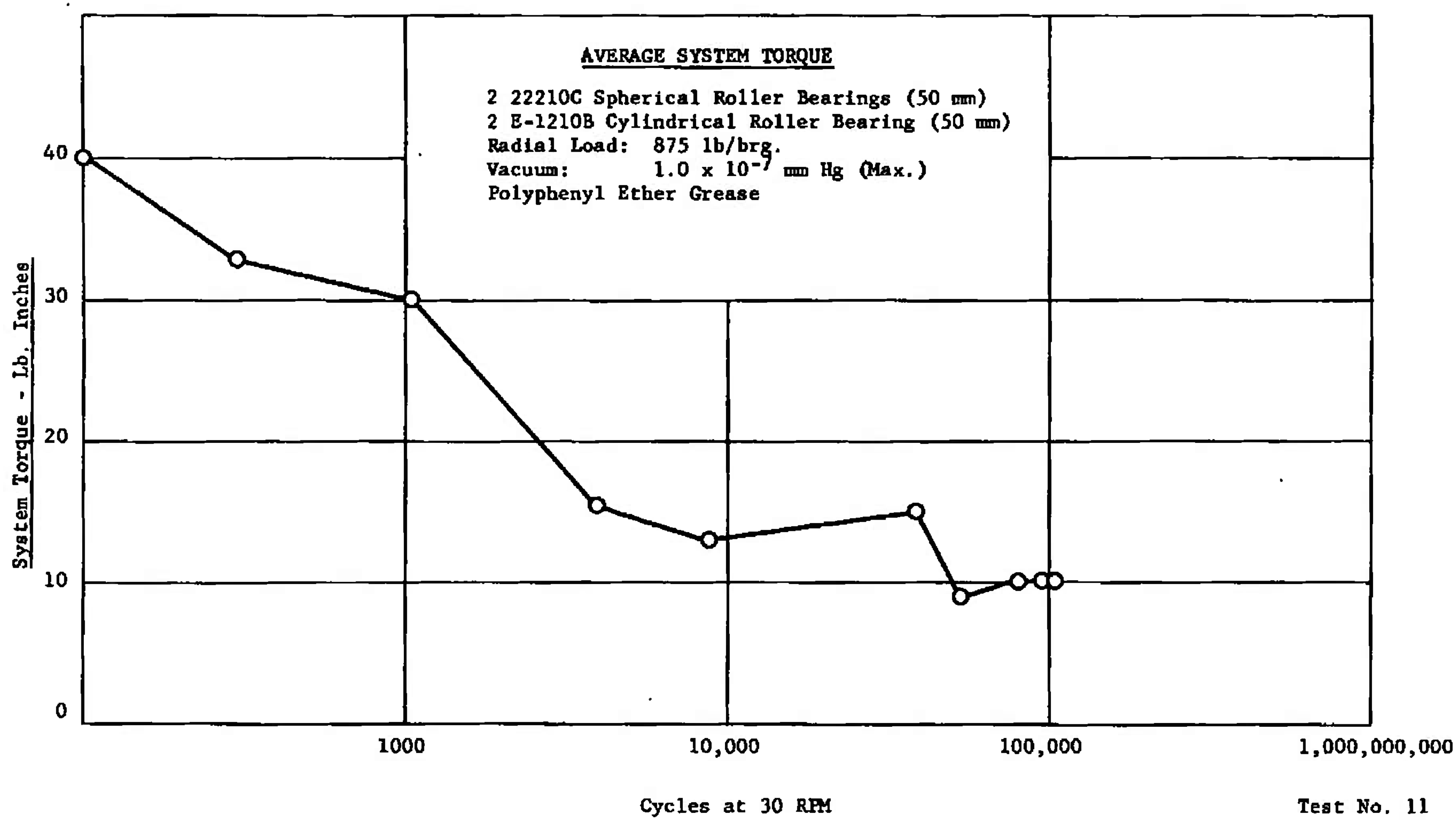

Fig. 27 Average System Torque Curves of the Various 50 mm Bearing-Lubricant Systems

TABLE 31

SMALL SCALE ROLLING ELEMENT (50 mm) TEST RESULTS

TEST NO.	BEARING SYSTEM Number Tested and Type	(1)* LUBRICANT	MAXIMUM RADIAL LOAD Lb/Brg.	(2) C_o/P	TOTAL RUN TIME (HRS) AIR (3)	TOTAL RUN TIME (HRS) VAC	TOTAL CYCLES AT 30 RPM AIR	TOTAL CYCLES AT 30 RPM VAC	VACUUM (MAX) mm Hg	DUTY (7) CYCLE %	BEARING SYSTEM TORQUE VS. CYCLES IN VAC.
1	3, 22210 Spherical	23 Kt Gold	1000	12	0.08	3.00	150	5400	3.0×10^{-7}	<1	See Fig.
2	4, 22210 "	24 Kt Gold	1200	10	0.16	2.52	300	4550	1.0×10^{-6}	<1	See Fig.
3	4, 22210 "	MoS_2 + Epoxy binder	1200	10	0.58	18.75	1050	33750	1.5×10^{-7}	3.2	See Fig.
4	4, 22210 "	MoS_2 + graphite + silicate binder	1000	10	1.28	31.55	2300	56850	1.2×10^{-7}	5.4	See Fig.
5	4, 22210 "	Methyl Chlorophenyl silicone (4)+ 5% MoS_2 low vapor pressure grease	1000	12	0.03	20.00	60	36000	4.3×10^{-7}	3.4	See Fig.
6	4, Timken Tapered Roller (5)	24 Kt Gold	--	--	--	--	--	--	--	--	--
7	4, Timken Tapered Roller	MoS_2 + Graphite + silicate	715	--	0.89	37.50	1600	66800	1.5×10^{-7}	6.4	See Fig.
8	2, 22210 Spherical 2, E-1210B Cyl.	Experimental MoS_2 + glass binder	875 875	13.8 8.6	0.33	52.50	600	94500	5.6×10^{-8}	9.0	See Fig.
9	4, Timken Tapered Roller	Silver plate	715	--	0.16	82.30	300	148200	1.0×10^{-7}	14.0	See Fig.
10	2, 22210 Spherical 2, E-1210B Cyl.	Petroleum base (6) low vapor pressure grease (Apiezon L)	875/1250	--	0.16	420.00	300	756000	4.0×10^{-8}	71.0	See Fig.
11	2, 22210C Spherical 2, E-1210B Cylindrical	Low vapor pressure polyphenyl ether base grease (experimental)	875	13.8 8.6	0.75	57.0	1350	102600	1.0×10^{-7}	12.0	See Fig.

* Footnotes follow

TABLE 31 FOOTNOTES

SMALL SCALE ROLLING ELEMENT (50mm) TEST RESULTS

1. Where solid film type of lubrication was used all bearing surfaces were coated with the exception of the rollers. Where used, the low vapor pressure greases were applied directly to the bearing metal surface.

2. C_o/P is the ratio of the static load capacity of the bearing to the applied load where C_o is defined as that radial load which will result in a permanent deformation of 0.0001 times the diameter of the rolling element. P is the applied load to the bearing.

3. The short period the bearings operated in air constituted a "run-in" period in which proper loading was accomplished as well as an initial screening of bearing performance prior to installation in the vacuum chamber.

4. Experimental low vapor pressure grease mixture consisting of G300 + 5% MoS_2.

5. Test failed in air before proper loading could be accomplished. High plating wear and flaking attributed to error in processing (cleaning preparation and/or plating process).

6. Low vapor pressure petroleum base grease called Apiezon L.

7. Duty Cycle (%) is based on 2 rpm continuous operation for 1 year and is discussed in detail in Section VII, "Evaluation Techniques".

8. Thermocouples positioned on the O.D. of two of the test bearings indicated operating temperature ranges for tests 1-10 as follows:

Test No.	Bearing OD - Temp. °F	Test No.	Bearing OD - Temp. °F
1	62 - 82	6	-------
2	74 - 81	7	65 - 79
3	75 - 83	8	64 - 90
4	68 - 86	9	66 - 79
5	62 - 99	10	71 - 93

TABLE 32

APPROXIMATE QUANTITIES OF LUBRICANT USED IN BEARING TEST NO'S 10 AND 11

BEARING TEST 10

Lubricant - Apiezon L (low vapor pressure petroleum distillate grease)

BEARING NO.(1)	APPROXIMATE AMOUNT OF LUBRICANT USED (gms)(2)
3R	2.1462
4R	3.7172
18S	4.9100
19S	5.5042

BEARING TEST 11

Lubricant - XSG-6158 (low vapor pressure polyphenyl ether base grease)

BEARING NO.	APPROXIMATE AMOUNT OF LUBRICANT USED (gms)
20S	3.9951
21S	4.5115
6R	2.4353
7R	2.8573

NOTE:

(1) Bearing numbers referred to are described in Table ,"Examination Results of ATL Small Scale Bearing Tests"

(2) The amount of lubricant used refers to the approximate amount of grease used in lubricating the bearing prior to testing.

TABLE 33

WEIGHT CHANGE ANALYSIS

A preliminary check was made on two low vapor pressure type greases. A known amount of grease of each type was placed in a carefully cleaned beaker, and a watch glass type dish, respectively. The greases were then subjected to a static 500 hour test in a vacuum of 1 x 10^{-7} mm Hg (max). The results are tabulated as follows:

Grease Type	Grease Trade Name	Initial Weight (gm) Beaker(1)	Dish(2)	Weight Change (gm) Beaker	Dish	% Wt Loss Beaker	Dish
Petroleum distillate	Apiezon L	3.9989	7.1456	0.0004	---(3)	0.01	--
Methyl Chlorophenyl Silicone grease + 5% Ag additive	G300 + 5% Ag(4)	4.2539	16.7091	-0.0423	-0.1881	1.00	1.12

Notes:

(1) Approximate surface area of grease exposed in beaker was 0.59 $in.^2$
(2) Approximate surface area of grease exposed in dish was 7.065 $in.^2$
(3) While there appeared to be little change in weight several foreign particles found on the grease surface tended to void this measurement.
(4) The 5% silver was added by ATL and is not a normal constituent of G-300 per se.

Oils of the above greases have reportedly been used successfully with lightly loaded ball bearings at vacuum levels in the order of 10^{-9} mm Hg for periods in excess of one year (*).

REFERENCES:

* Proceedings of Space Lubrication Conference FL 210/53, Sept. 1963, published by Fuels and Lubricants Division, Office of the Director of Defense Research and Engineering.

TABLE 34

EXAMINATION RESULTS OF ATL SMALL SCALE BEARING TESTS (50 mm)

SPHERICAL ROLLER BEARING SKF 22210C (50 MM)	SPHERICAL ROLLER BEARING SKF 22210C (50 MM)	SPHERICAL ROLLER BEARING SKF 22210C (50 MM)	SPHERICAL ROLLER BEARING SKF 22210C (50 MM)
Test No. 1 Lub - 23 Kt Au Envir. - Vac (3×10^{-7} mm Hg Max.)			
ROLLERS-Highly polished. No evidence of surface damage	ROLLERS-Highly Polished. No evidence of surface damage	ROLLERS-Highly Polished. No Damage	*
OR-Load Zone Polished, light flaking-coating intact-some discoloration.	OR-Highly polished, especially over load zone-occasional streak of dark film (easily removed) over gold plate-coating intact.	OR-Highly Polished in load zone-same as 2S	
IR-Polished-Band of light pickup-coating intact-some discoloration.	IR-Highly polished-light pickup along one edge.	IR-Highly polished-coating intact.	
CAGES-Polished over contact points-no damage.	CAGES-Contact areas polished-wear nil.	CAGES-Contact areas polished-wear nil.	
RR-Lightly burnished	RR-Lightly Burnished	RR-Lightly burnished	
Brg. No. 1S	Brg. No. 2S	Brg. No. 3S	
Test No. 2 Lub - 24 Kt Au Envir - Vac. (1×10^{-6} mm Hg Max)			
ROLLERS-Polished-no damage	ROLLERS-Polished-no damage	ROLLERS-Polished-no damage to surface	ROLLERS-Highly Polished-excellent condition
OR-Highly polished-several thin bands of dark deposit that is easily scraped off-coating intact	OR-Highly polished over load zone-coating intact	OR-Highly polished-coating intact	OR-Polished-coating intact

NOTE:
OR = outer race IR = inner race Cage or Retainer RR = Retainer Ring
* Three bearing test.

TABLE 34 (Continued)

SPHERICAL ROLLER BEARING SKF 22210C (50 MM)	SPHERICAL ROLLER BEARING SKF 22210C (50 MM)	SPHERICAL ROLLER BEARING SKF 22210C (50 MM)	SPHERICAL ROLLER BEARING SKF 22210C (50 MM)
	Test No. 2 contd.		
IR-Highly polished-some pickup with several areas built-up considerably .	IR-Polished-several dark bands.	IR-Polished-some pickup on edge of race-coating intact.	IR-polished-coating intact.
CAGES-Polished over contact area.	CAGES-Polished over contact points.	CAGES-Polished over contact points-wear nil.	CAGES-Wear nil-polished over contact points.
Brg. turns freely 12S	Brg. turns freely 13S	Brg. turns freely 14S	Brg. turns freely 15S
	Test No. 3 Lub - MoS_2 + Epoxy binder (106) Envir - Vac (1.5×10^{-7} mm Hg max)		
ROLLERS-Polished and discolored circumferential bands. Discoloration mainly from solid film-some rollers show straw colored bands from retainer wear. No significant damage to rollers.	ROLLERS-Partially polished and partially discolored. Light pickup of coating on discolored areas-no roller damage.	ROLLERS-Same as 8S	ROLLERS-Polished-some discoloration bands primarily from coating pickup-no damage.
OR-Coating polished-Lub depleted in some areas as evidenced by metal exposure.	OR-Coating still effective although considerably worn and marginal in spots-polished.	OR-Coating partially intact in one area and depleted in another.	OR-Polished-coating flaky in some areas-Lub appears marginal in other areas.
IR-Coating polished-smooth circumferential band approx. 1/8" wide resulting from cage wear (one side only)	IR-Coating flaky and marginal in spots.	IR-Surface smooth and discolored. Possibly a combination of coating remaining & heat effects.	IR-Discolored (Purple to Black bands)-coating flaky.

TABLE 34 (Continued)

SPHERICAL ROLLER BEARING SKF 22210C (50 MM)	SPHERICAL ROLLER BEARING SKF 22210C (50 MM)	SPHERICAL ROLLER BEARING SKF 22210C (50 MM)	SPHERICAL ROLLER BEARING SKF 22210C (50 MM)
	Test No. 3 (Contd)		
CAGES-Light to moderate wear where coating was removed.	CAGES-Coating intact-no cage wear.	CAGES-Evidence of moderate galling, pickup & wear.	CAGES-Coating intact generally Light wear in a few pockets.
RR-Coating polished & intact.	RR-Coating intact.	RR-Coating Buffed.	RR-Flaky but intact.
4S	8S	5S	6S
	General Comments - coating marginal on four bearings		
	Test No. 4 Lub - MoS_2 + Graphite + Silicate Envir. - Vac (1.2×10^{-7} mm Hg Max)		
ROLLERS-Smooth, polished & Discolored-no damage	ROLLERS-Polished-some coating transferred to rollers resulting in gray color bands-no damage.	ROLLERS-Polished-considerable coating pickup-no damage to roller.	ROLLERS-Polished-some coating pickup-no damage to rollers.
OR-Polished in contact area. Surface discolored and lub marginal-no race damage.	OR-Coating Marginal in load areas-no race damage.	OR-Coating marginal in load areas-Polished.	OR-Coating marginal-polished-no race damage.
IR-Polished & Discolored-No damage-Lub marginal	IR-Polished-coating marginal in some areas-Burnished & intact in others.	IR-Contact area polished-coating flaky in some areas-coating marginal.	IR-Same as OR
CAGES-Light to moderate cage wear.	CAGES-Coating generally intact.	CAGES-Coating intact-wear nil.	CAGES-Coating intact-wear nil.
RR-Coating intact.	RR-Coating intact.	RR-Coating intact.	RR-Coating intact.
Brg. Rotates freely 7S	Brg. Turns freely 9S	Brg. grabs when rotating because of excessive coating pickup on rollers. 10S	Rotates freely. 11S

General Comments:

While the coating had worn considerably based on its initial 0.0002-0.0005 in. thickness, none of the bearing components were damaged. The polished appearance of the races (semi-metallic luster) indicated that remaining lubricant was partially effective either by virtue of a remaining monolayer or by replenishment from the adjacent areas where the coating was considerably intact.

TABLE 34 (Continued)

SPHERICAL ROLLER BEARING SKF 22210C (50MM)	SPHERICAL ROLLER BEARING SKF 22210C (50 MM)	SPHERICAL ROLLER BEARING SKF 22210C (50 MM)	SPHERICAL ROLLER BEARING SKF 22210C (50 MM)
Test No. 5 Lub - Grease (G30C + 5% MoS_2) Exp. Envir - Vac (4.3×10^{-7} mm Hg Max.)			
ROLLERS-Smooth, Polished, Evidence of Pickup on some roller surfaces on a microscopic scale.	ROLLERS-Smooth, Polished, Thin light band around some rollers-Surfaces in good condition in general.	ROLLERS-Gray discoloration on surface due primarily to thin film of MoS_2-Some show light orange discoloration.	ROLLERS-Smooth, moderately discolored-some roller surfaces glassy and granular in appearance.
OR-Moderate pickup and galling in load zone.	OR-Smooth, burnished band in load zone-surface appearance good.	OR-Smooth, burnished band in load zone-no damage.	OR-Generally smooth-one band in load zone appears discolored-is glassy and granular in appearance.
IR-1/16" wide band adjacent to cage spacer where pickup & galling occurred.	IR-Light burnishing over roller path-no damage-Light pickup where cage separator spacer ring rests.	IR-Roller path on one side of race near outer periphery granular in appearance.	IR-Same as OR except two bands
CAGES-A few pockets indicate light wear, the rest no wear.	CAGES-A few pockets indicate wear-the majority of pockets indicate no wear.	CAGES-A few pockets indicate wear-majority of pockets indicate no wear.	CAGES-Extremely light wear.
SSR-Light wear on sides.	SSR-Light wear on sides and ID of ring.	SSR-Light scuffing on one side.	
Bearing rotates quietly but was stiff.			The type of granular appearance noted on the races was typical of a marginal lubrication condition. Sufficient grease lub was apparently lacking.
Brg. No. 21S	Brg. No. 22S	Brg. No. 23S	Brg. No. 24S

*NOTE: OR = outer race IR = inner race Cage or Retainer SSR = Separator Spacer Ring (Separates Split Type Cage)

TABLE 34 (Continued)

TIMKEN TAPERED ROLLER BEARING(1) (50 MM)	TIMKEN TAPERED ROLLER BEARING (50 MM)	TIMKEN TAPERED ROLLER BEARING (50 MM)	TIMKEN TAPERED ROLLER BEARING (50MM)
Test No. 7 - Lub - MoS_2 + Graphite + Silicate - Envir - Vac (1.5×10^{-7} mm Hg Max.)			
ROLLERS-Thin buildup of coating transferred from races to contact areas of roller surfaces. No damage to base metal.	ROLLERS-Same as "1T".	ROLLERS-Light gray discoloration on roller surfaces from coating-no buildup-smooth surface.	ROLLERS-Light Gray discoloration-burnished appearance-surface slightly rough upon scratching.
OR-Coating marginal. In part of load zone, coating removed leaving highly polished metal surface.	OR-Coating marginal in load zone-thin film observed over polished load zone area. Coating intact on remainder of race surface.	OR-Same as "2T"	OR-Same as "2T"
IR-Most of initial coating thickness removed-surface gray and polished-extremely thin film of coating noted on surface along with particles of coating buildup.	IR - Same as "1T"	IR-Excellent condition, however less gray discoloration (from coating) observed & more polished metal exposed.	IR-Same as "1T" only coating buildup nil.
CAGE-Light polishing-wear nil - Coating intact.	CAGE-Same as "1T"	CAGE-Light polishing of coating-wear nil - coating intact.	CAGE - Light polishing-some light wear indicated in several pockets - coating mostly intact.
Brg. Turned Freely	Brg. Turned Freely	Brg. Turned Freely	Brg. Turned Freely
Brg. No. 1T	Brg. No. 2T	Brg. No. 3T	Brg. No. 4T

Note

(1) Timken Tapered Roller Bearing, Cone No. 365, Cup No. 363

TABLE 34 (Continued)

CYLINDRICAL ROLLER BEARING ROLLWAY EB 1210 (50MM)	SPHERICAL ROLLER BEARING SKF 22210C (50 MM)	SPHERICAL ROLLER BEARING SKF 22210C (50 MM)	CYLINDRICAL ROLLER BEARING ROLLWAY EB 1210 (50 MM)
Test No. 8 Lub - Experimental Glass + MoS_2 Envir - Vac (5.6×10^{-8} mm Hg Max)			
ROLLERS-Polished, burnished-at 27X Mag. appearance is somewhat similar to that of a vapor blasted surface.	ROLLERS-Polished, glassy & shiny in appearance as if thin film is covering roller surface. Light buildup of brass flakes on some rollers-general surface condition good.	ROLLERS-Highly polished and glassy-occasional small flake of easily removable solid film coating-Light straw colored bands-surface appearance good.	ROLLERS-Polished-some dark bands-glassy surface giving appearance of thin film.
OR-Surface smooth, coating appears depleted in roller path-evidence of iron oxide-no apparent damage.	OR-Major portion of coating still intact-some areas highly polished-area adjacent to separator spacer ring indicates some redish debris on surface, coating intact underneath however-also some flaking where coating has been builtup. No evidence of base metal damage.	OR-Thin Film over majority of load area-some brass pickup which ultimately became a thin film due to rolling action-Redish debris adjacent to separator spacer.	OR-Coating appears depleted in roller path although thin film evidenced outside of load zone. No base metal damage.
IR-Coating depleted over roller path-smooth burnished appearance.	IR-Coating highly effective on one side of race, on other roller track a uniform coating of Brass (From the retainer wear) is effectively protecting the base metal surface*	IR-Polished-thin effective film remaining on one side of inner race-other track indicates thin coating of brass. Some thin flakes of brass also observed-no evidence of damage to base metal.	IR-Glassy surface appearance over roller path-smooth uniform surface-major portion of coating removed. No base metal damage.
CAGE-Cage wear.	CAGES-One split cage-no wear-the other cage high wear.	CAGES-High wear and galling on one split cage-No wear on the other.	CAGE-Moderate cage wear.
SR-Some coating buildup-coating still effective.	SSR-Coating intact.	SSR-Coating intact.	SR-Coating effective.
Brg. No. 1R Lub-Binder Ratio 1:1	Brg. No. 16S Lub-Binder Ratio 1:1	Brg. No. 17S Lub-Binder Ratio 2:1	Brg. No. 2R Lub-Binder Ratio 2:1

*Note-Brass wear particles from retainer wear transferring to roller tracks of the inner race is not a desirable situation; however the particles were deformed to a degree where the brass became an effective coating with good attachment to the glass coating and/or base metal.

TABLE 34 (Continued)

TIMKEN TAPERED ROLLER BEARING (50 MM)	TIMKEN TAPERED ROLLER BEARING (50 MM)	TIMKEN TAPERED ROLLER BEARING (50 MM)	TIMKEN TAPERED ROLLER BEARING (50 MM)
Test 9 Lub - Ag Plate Envir - Vac (1.0×10^{-7} mm Hg Max)			
ROLLERS-Polished-No damage.	ROLLERS-Polished-No damage.	ROLLERS-Polished-No damage.	ROLLERS-Polished-No damage.
OR-Coating intact-although worn and polished in load area.	OR-Same as "7T"	OR-Highly polished-coating worn in load area.	OR-Polished-small area in load zone where coating appears removed, Ag blistered in area adjacent to it-possibly due to heating effect or poor bond.
IR-Coating intact-polished-excellent condition.	IR-Same as "7T"	IR-Highly polished-coating worn-thin film still effective.	IR-Polished-High coating wear in 3/16" wide band around race circumference marginal lubrication to exposure of base metal within band. Band smooth and polished.
CAGE-Polished where rollers made contact-good condition.	CAGE-Same as "7T"	CAGE-Some light wear - otherwise same as "7T"	CAGE-Same as "7T"
Brg. Turned Freely Brg. No. 7T	Brg. Turned Freely Brg. No. 8T	Brg. Turned Freely Brg. No. 9T	Brg. Turned Freely Brg. No. 10T

TABLE 34 (Continued)

CYLINDRICAL ROLLER BEARING ROLLWAY EB 1210 (50MM)	SPHERICAL ROLLER BEARING SKF 22210CY (50MM)	SPHERICAL ROLLER BEARING SKF 22210CY (50MM)	CYLINDRICAL ROLLER BEARING ROLLWAY EB 1210 (50 MM)
Test 10 Lub - Low Vapor Petroleum Distillate (Apiezon L) Envir - Vac (5 x 10^{-8} mm Hg Max.)			
ROLLERS-Polished-Smooth, excellent condition.	ROLLERS-Polished-Smooth-Some moderately spalled areas on some roller surfaces.	ROLLERS-Polished-Smooth-Excellent condition.	ROLLERS-Highly polished-Smooth Excellent condition.
OR-Polished-Smooth, excellent condition.	OR-Generally polished and smooth-spalled band (moderate) approx. 1/16" wide in load zone.	OR-Same as Bearing 3R.	OR-Same as Bearing 3R.
IR-Polished-Smooth, excellent condition.	IR-Same type of band appears on one of the two existing IR race surfaces. Implies band may have been caused by some unusual load condition.	IR-Same as Bearing 3R.	IR-Same as Bearing 3R.
CAGE-Wear nil-lightly burnished in some contact areas-excellent condition.	CAGES-Wear nil-some burnishing in contact areas.	CAGES-Light wear on several cage pockets-The majority of pockets had no wear.	CAGES-No wear-Excellent condition.
SR-No wear.	SSR-Light burnishing-wear nil.	SSR-Light burnishing.	SR-Polished in some areas
Bearing turns freely Brg. No. 3R	Bearing turns freely Brg. No. 18S	Bearing turns freely Brg. No. 19S	Bearing turns freely Brg. No. 4R

FOOTNOTE:

1. Spalling-The term spalled as used here implies a pitting or flaking action at the surface, generally on a microscopic scale. The cause in the case of bearing 18S appears to be attributed to excessive loading over a specific area.

TABLE 34 (Continued)

CYLINDRICAL ROLLER BEARING ROLLWAY EB 1210 (50MM)	SPHERICAL ROLLER BEARING SKF 22210CY (50MM)	SPHERICAL ROLLER BEARING SKF 22210CY (50MM)	CYLINDRICAL ROLLER BEARING ROLLWAY EB 1210 (50MM)
Test 11 Lub - Low Vapor Pressure Polyphenyl Ether Base Grease Envir - Vac (1.2 x 10^{-7} mm Hg Max)			
ROLLERS-Polished-Light brown discoloration band located at one end of rollers more of a combination of burnishing and lubricant effect-No apparent damage.	ROLLERS-Highly Polished-Good condition. No film deposit.	ROLLERS-Highly polished-Excellent condition.	ROLLERS-Same as 6R. Easily removable lacquer-like film (from Lub) deposited on some roller surfaces. No surface damage.
OR-Lightly burnished-wear nil. No surface damage. Good condition.	OR-Excellent condition-polished.	OR-Same as 20S.	OR-Same as 6R.
IR-Light burnished band approx. 1/8" wide. Brown lacquer-like film on some areas of band easily removable. No damage to surface.	IR-Polished-excellent condition.	IR-Same as 20S.	IR-Same as 6R.
CAGE-Wear nil-excellent condition.	CAGES-Light wear in several pockets-The majority of pockets incurred no wear.	CAGES-Same as 20S.	CAGE-Lightly burnished-Good condition.
SR-Lightly burnished. Good condition.	SSB-Good condition. No detectable wear or surface damage.	SSB-Same as 20S.	SR-Same as 6R.
Bearing Turns Freely. Brg. No. 6R	Bearing Turns Freely. Brg. No. 20S	Bearing Turns Freely. Brg. No. 21S	Bearing Turns Freely. Brg. No. 7R

C. GEAR TEST APPARATUS

The gear tests were conducted in a Four-Square Gear Test Apparatus which consisted of two Model H-1270 Boston Gear Box units coupled together to form a four-square arrangement. With this arrangement two sets of gears could be tested at the same time.

The gear ratio was 3.2:1 with the large helical gear possessing a P.D. of 4 inches and the pinion having a P.D. of 1.25 inches. The gear support bearings consisted of Timken tapered roller bearings. Ball bearings were used in the initial tests but were soon replaced with tapered rollers because of the former's poor performance.

The large 4 inch P.D. gears were joined together through a flexible coupling. The pinion gears were coupled by means of a specially designed split metal coupling through which, by means of a torsional force, a load could be introduced on all of the test gears. The load was introduced in the following manner. One half of the split coupling that was attached to a pinion shaft was held rigid. The other half, also attached to a pinion shaft, was slowly rotated (torsional force) until the desired load was applied. Both cylindrical split couplings were slotted so that once the desired load was reached the couplings could be held in place by bolts positioned through the slots. The half of the coupling to which the torsional load was applied also incorporated a 3/8-inch diameter torsion rod instrumented with strain gages which were calibrated to measure the load applied to the gears.

Figure 28 is a photograph of the Four-Square Gear Tester which shows the design features discussed quite well. The apparatus is shown in its test position as it would be in the vacuum chamber, and as one may note the tops of the gear housings have been machined off and the back support plate slotted. This was done to insure proper gear exposure to vacuum for lubricant testing purposes.

To further insure that the machined gear housing openings were adequate for the purpose stated, a brief study was made and is reported in this section under the heading, "Gear Tester - Vacuum Study".

<u>Gear Tester - Vacuum Study</u>

A brief study was undertaken to determine the adequacy of the gear test modified housing openings to insure proper gear exposure to vacuum for testing purposes.

The small pressure differential calculated between the chamber pressure and pressure within the gear box was considered to have little effect on the gear tests either in the ATL chamber or when used in the larger Arnold Aerospace Chamber. The results of this study are described below:

The assumed boundary conditions were:

1. The pumping for the vacuum chamber is approximately 700 liters/sec. for air type gases. The diffusion pump with a basic speed of 1440 liters/sec

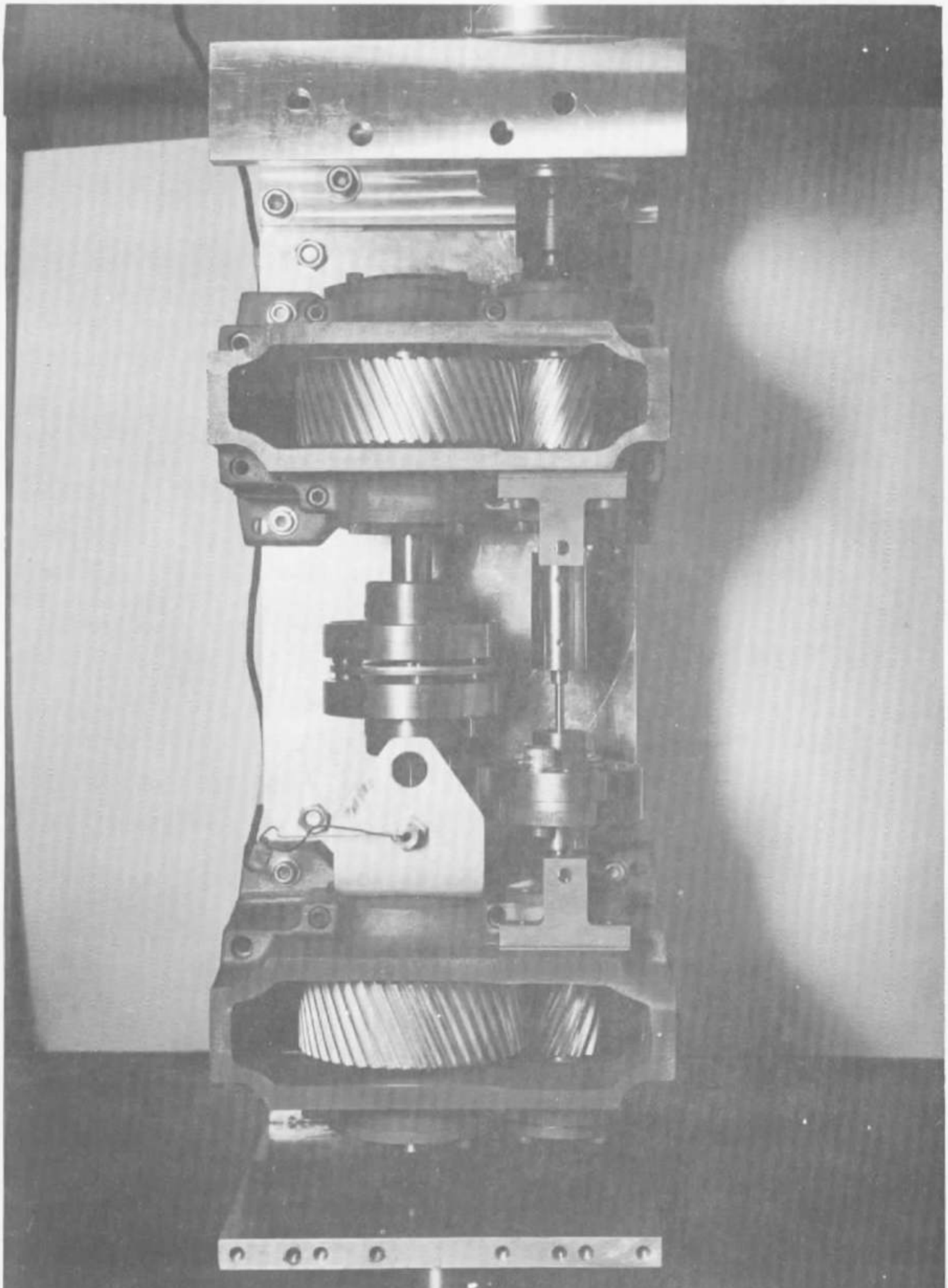

Fig. 28 Four-Square Gear Tester

has a single bounce water trap, and a single bounce liquid nitrogen trap which are an integral part of the system and cut the speed by about a factor of 2.

2. The ultimate pressure of the empty chamber is about 5×10^{-9} torr which leads to an inherent outgassing or leak input of about 3×10^{-6} torr liters/sec.

3. The ultimate pressure with open trap gear boxes in the chamber is approximately 1×10^{-7} torr which gives an outgassing rate of 7×10^{-5} torr liters/sec. This indicates that the chamber outgassing can be neglected in further calculations.

With this input data and making the worst case assumption that the outgassing comes only from the interior of the gear boxes, and assuming that the two gear boxes are identical, we can arrive at the maximum pressure increment of the interior of the box above the pressure in the chamber.

$ZQ = P - P_p$ where Z = The impedance of the orifice between geared case and chamber

Q = The throughput of the orifice = 3.5×10^{-5} liters/sec

P = Pressure in the gear case

P_p = Pressure in the chamber - 1×10^{-7} torr

$Z = \frac{1}{C}$ where C = Conductance of the orifice. 300 K air = 1300 liters/sec.

$P = 1.3 \times 10^{-7}$ torr

From this we conclude that in the case where the effusion of the gas from the gear case is limited by the orifice, the maximum pressure of the gear case above that of the chamber is 3×10^{-8} torr. In this particular set of experiments it seems improbable that this small increment will have appreciable effect.

Testing Criteria

Monitoring of gear performance for each test was accomplished in much the same manner as the bearing tests with the exception of measuring torque throughout the operating period. The Four-Square Gear Test Apparatus was driven through a magnetic clutch system in a manner similar to that of the Bearing Tester. Clutch slippage could occur around 42 lb-in. thereby acting as a fuse for the prevention of any catastrophic failures.

Speed monitoring was another means used to detect the first signs of lubricant wear, transfer, and buildup. The speed would become erratic, for example, when sufficient buildup of lubrication on the gear teeth had occurred. Some tests were visually inspected several times throughout the test for evidence of coating wear.

Varying degrees of wear, transfer and buildup occurred throughout the small scale gear tests, however none of the gears tested experienced a seizure type of failure. Further, all of the coated gear tests showed at their termination that they could have been operated over a substantial longer period. Test periods varied depending upon the testing purpose for a particular type of lubricant and in some cases testing schedule.

D. SMALL SCALE GEAR TESTS RESULTS

Four-Square Gear Tests - Platings

Tables 35 and 36 summarize the results of four gear tests using their platings of 23 kt gold and silver. Two tests were run in air, and two in vacuum, both over the same operating period for purposes of comparison.

1. In Air

   Macroscopic and microscopic observations indicated that the two sets of test gears could be operated over a considerably longer period than the 78 hours tested, based on the general condition of the gear teeth. Limited material transfer (moderate galling), and evidence of small areas of coating removal (further confirmed by small amounts of wear debris found to be magnetic) would not necessarily prevent the gears from functioning unless the condition became progressively worse in a short period of time.

2. In Vacuum

   The extent of material transfer and coating removal in vacuum (9.8 x 10^{-8} mm of Hg) over a 78 hour period was found to be considerably less than that observed on the gear teeth surfaces in air for the gold and silver coatings. The gears were capable of continued operation as the coatings were intact.

Four-Square Gear Tests - Organic Coatings

Helical gears with solid film lubricants consisting of (1) MoS_2 + graphite + silicate binder, and (2) MoS_2 + epoxy binder were successfully run for 230 hours at a surface compressive stress level of approximately 30,000 psi, in a vacuum measured to 6.5 x 10^{-8} mm Hg (max.). The ratio between the pinion and large helical gear was 3.2:1, resulting in a 80 rpm pinion gear speed and a 25 rpm large helical gear speed. The tests results are reported in Table 37. These tests were terminated in 230 hours to allow for testing other coatings. The gear coatings were found in excellent condition and could have sustained considerably more hours of operation.

The Timken tapered gear support bearings coated with the same solid film lubricants also performed satisfactorily in vacuum.

TABLE 35

ATL FOUR-SQUARE GEAR TEST RESULTS – PLATINGS

TYPE: Helical Gears
SIZE: 4.00 P.D. (Meehenite Material)
1.25 P.D. Pinion (4140 Material)

SPEED: 25 RPM / 80 RPM 3.2:1 Ratio
LOAD: 18,500 psi (compressive stress)

TEST NO. (1)	TEST PERIOD (HRS.)	GEAR COATING COMBINATION (6)		TORQUE (2) AT NO LOAD (OZ-IN)		TORQUE (2) AT PRELOAD (3) (OZ-IN)		ENVIRONMENT	BACKLASH (4) MILS		COMMENTS
		PINION	LARGE	BREAKAWAY	RUNNING	BREAKAWAY	RUNNING		TOP GEAR SET	BOTTOM GEAR SET	
1	Initial	Ag	Au	3.5	3.5	12.5	12.5	Air - R.T.	15.90		
	78 Hrs.			4.5	4.0	16.0	14.0	"	17.71		Test suspended at first indication of coating removal. Gears still capable of operation
2	Initial	Au 23 Kt	Ag					"		11.34	
	78 Hrs.							"		11.71	
3	Initial	Ag	Au	3.75	3.0	20.0	16.0	9.8×10^{-8} mm of Hg R.T.	16.71		
	78 Hrs.			3.5	3.0	21.0	17.0	"	16.73		Test suspended in 78 hrs. to compare with air test. Coatings still intact and capable of longer operation.
4	Initial	Au 23 Kt	Ag							11.98	
	78 Hours									12.46	

FOOTNOTES:

(1) The Four-Square Gear Tester accommodates two tests simultaneously, i.e. No. 1 and 2, etc.
(2) System torque, i.e. includes both sets of test gears and support bearings - Measurements taken at input shaft.
(3) 12 in-lb preload (corresponds to 18,500 psi tooth surface compressive stress)
(4) Backlash is a measurement of the difference between the thickness of a tooth and the width of space in which it meshes
(5) Support bearings - for pinion helical gear - Timken rollers, gold plated - thin film of Apiezon L
- for large helical gear - Norma Hoffman ball bearings - thin Film of Apiezon L
(6) Plating thickness was 0.00015-0.00025 inches. A copper flash preceded the application of both coatings.

TABLE 36

WEIGHT CHANGE OF INDIVIDUAL GEARS TESTED IN TABLE 35 (VACUUM TEST)

Gear and Coating	Weights of Gears With Plating Weight (initial) (Grams)	Weight After 78 hrs. in Vac. (Grams)	Loss or Gain (Grams)
1) Large Gold Gear	1586.03	1586.02	-.01
2) Large Silver Gear	1793.36	1793.33	-.03
3) Small Gold Pinion Gear	396.715	396.711	-.004
4) Small Silver Pinion Gear	430.820	430.810	-.01

4.00" P.D. Helical Gear Meehanite

1.25" P.D. Helical Pinion Gear 4140

TABLE 37

ATL FOUR-SQUARE GEAR TEST RESULTS – COATINGS

Type: Helical Gears
Size: 4.00" P.D. (Meehenite Matl)
1.25" P.D. Pinion (AISI 4140 Matl)

SPEED: 80:25 RPM 3.2:1 ratio
LOAD: 30,000 psi (compressive stress)

TEST NO (1)	TEST PERIOD (HRS.)	GEAR COATING COMBINATION		TORQUE (2) AT NO LOAD (OZ-IN)		TORQUE (2) AT PRELOAD (3) (OZ-IN)		ENVIRONMENT	BACKLASH (4) MILS		COMMENTS
		PINION	LARGE	BREAKAWAY	RUNNING	BREAKAWAY	RUNNING		TOP GEAR SET	BOTTOM GEAR SET	
1	Initial	Graphite plus MoS_2 plus Silicate Binder	Graphite plus MoS_2 plus Silicate Binder	40	28	108	88	6.5×10^{-8} mm Hg Max at R.T.	--		Coating intact & Burnished. Moderate buildup in some areas. Gears capable of operation many more hours.
	After 230 hrs.	"	"	2.0	1.5	32	28	"	18.4		
2	Initial	MoS_2 + Epoxy	MoS_2 + Epoxy					6.5×10^{-8} mm Hg Max at R.T.		10.4	Coating intact relatively uniform & polished Gears capable of operation many more hours.
	After 230 hrs.	"	"					"		13.0	

FOOTNOTES:
(1) The Four-Square Gear Tester accommodates two tests simultaneously, ie, No. 1 and 2
(2) System torque, i.e. includes both sets of test gears and support bearings-Measurements taken at input shaft
(3) 30 in-lb preload (corresponds to approx. 30,000 psi tooth surface compressive stress)
(4) Backlash is a measurement of the difference between the thickness of a tooth and the width of space in which it meshes
(5) Support bearings - Timken Tapered Roller for Test 1, graphite-MoS_2-Silicate coated
Timken Tapered Roller for Test 2, MoS_2 - Epoxy coated

Discussion of Test Results Reported in Tables 37 and 38

The overall appearance of the tested coatings described in Table 37 was different, the MoS_2-epoxy type looking better from the standpoint of good uniformity of coating, small amount of coating buildup, and considerable polishing of the coating. Both types of coatings were effective, however, and intact. The weight loss shown in Table 38 indicated that the MoS_2-graphite-silicate binder coating did undergo a greater weight loss, which may be of considerable significance from the standpoint of bonding effectiveness. On the other hand, the MoS_2-graphite-silicate does not apply as easily to a surface as the MoS_2-epoxy type, thus the former tends to end up as a thicker coating upon initial application. This excess thickness is generally removed during run-in or some subsequent period thereafter leaving a hoped-for thin, but effective film remaining. The disadvantage of the latter situation is the considerable amount of coating wear debris that is generated until a final, but effective thin coating remains that can support the loads involved. If the debris drops out or is pushed out of the gear or bearing component, smooth performance results. If it builds up, erratic torque conditions result. Another factor responsible for the generation of wear debris is that the MoS_2-graphite-silicate is a harder coating than the MoS_2-epoxy type; hence it possesses a somewhat brittle structure which contributes to what may be termed as a "top layer crust removal" effect during initial sliding. The magnitude of this effect will of course depend upon the initial thickness of the coating.

It is questionable as to what one may interpret as "marginal" lubrication when considering some types of solid film lubricants. The fact that a coated surface, after undergoing successful sliding, loses most of its original coating thickness does not necessarily indicate that the remaining coating, although extremely thin, is ineffective. Performance results on these bearings as well as gears would indicate, in some cases, that a thin film, probably in the order of angstroms and close to invisible to the naked eye, is effective in lubricating, as evidenced by the lack of scoring, galling or wear on the bearing or gear base metal surfaces upon subsequent microscopic examination. The extent of effectiveness of this thin monolayer film will be determined by the load, speed, and environmental parameters under which it has to perform.

Four-Square Gear Tests - Low Vapor Pressure Greases

Several special greases were tested with the gears in vacuum for comparison with solid film type lubricants primarily, and secondly, because of ATL's opinion that low vapor pressure greases offer considerable potential towards meeting high load, low velocity, gear and bearing requirements in high vacuum, especially in those cases where long operating life in the neighborhood of one year is required.

The greases consisted of (1) a petroleum distillate base grease commercially known as Apiezon L, and (2) a methyl chlorophenyl silicone grease commercially known as G-300. To the G-300 was added 3 percent silver.

TABLE 38

WEIGHT LOSS OF GEARS TESTED IN TABLE 37

Test No.	Helical Gear Combination	Coating	Weight (grams) Initial	Weight (grams) Final	Weight (grams) Loss
1	4.00" P.D. Gear	MoS_2-graphite silicate binder	1201.00	1200.860	0.240
	1.25" P.D. Pinion	" "	430.795	430.637	0.158
2	4.00" P.D. Gear	MoS_2-Epoxy binder	1185.280	1185.250	0.030
	1.25" P.D. Pinion	" "	391.609	391.594	0.015

The results of the Four-Square Gear Tests can be seen in Table 39

A recorded uniform speed, and low noise level tended to indicate that the gears ran satisfactorily for the desired 500 hour period in the ATL Vacuum Chamber. While operating performance appeared satisfactory, subsequent examination revealed that high wear had taken place as well as considerable galling on the gear teeth contact surfaces. Surprisingly, the magnitude of galling was not of a varying nature but had taken place fairly uniformly over those areas that had been subjected to high wear. Perhaps this accounted for the relatively uniform torque and speed noted throughout the test.

These results, while limited, indicate that to improve gear performance the gear materials should be considerably harder, and the greases must undergo further development, primarily towards improving their load carrying capabilities for purposes of meeting highly loaded low velocity gear as well as bearing requirements in space applications.

TABLE 39

ATL FOUR SQUARE GEAR TEST RESULTS – LOW VAPOR GREASES

Type: Helical Gears
Size: 4.00" P.D. (Meehanite Matl)
1.25" P.D. Pinion (AISI 4140 Matl)

SPEED: 80:25 3.2:1 ratio
LOAD: 30,000 psi (compressive stress)

TEST NO. (1)	TEST PERIOD (HRS)	GEAR COATING COMBINATION PINION	LARGE	TORQUE (2) AT NO LOAD (OZ-IN) BREAKAWAY	RUNNING	TORQUE (2) AT PRELOAD (3) (OZ-IN) BREAKAWAY	RUNNING	ENVIRONMENT	COMMENTS
1	Initial	Low vapor pressure petroleum distillate grease (Apiezon L)	Low vapor pressure petroleum distillate grease (Apiezon L)	13	---	40	--	1 x 10^{-7} mm Hg Max at R.T.	High tooth wear and galling observed. Still operational.
	After 500 hrs.	"	"	6	6	24	22	"	
2	Initial	Low vapor pressure methyl chlorophenyl silicone grease (G-300) + 3% Ag additive.	Low vapor pressure methyl chlorophenyl silicone grease (G-300) + 3% Ag additive.					1 x 10^{-7} mm Hg Max at R.T.	High tooth wear and galling observed. Still operational.
	After 500 hrs.	"	"					"	

FOOTNOTES:
(1) The four-square gear tester accommodates two tests simultaneously, i.e., No. 1 and 2.
(2) System torque, i.e. includes both sets of test gears and support bearings-Measurements taken in input shaft.
(3) 30 in-lb preload (corresponds to approx. 30,000 psi tooth surface compressive stress)
(4) Support bearings - Timken Tapered Roller for Test 1, Low vapor pressure petroleum distillate grease (Apiezon L)
Timken Tapered Roller for Test 2, Low vapor pressure methyl chlorophenyl silicone grease (G-300) + 3% Ag additive.
(5) Approximate amounts of grease applied to gears: Top gear set - 6.06 gms - Bottom gear set - 7.38 gms

SECTION X - LARGE SCALE BEARING TESTS (100 mm)

A. TEST FACILITY AND EQUIPMENT

The Seven-Foot Aerospace Research Chamber at AEDC was used to provide the test environment for the Advanced Technology Laboratories' large scale tests. The inside dimension of the chamber is 7 foot diameter by 12 foot from door flange to door flange. A view of the inside of the chamber with the bearing test apparatus in the foregound and the gear test apparatus in the background can be seen in Figure 29.

The pumping system for the chamber consisted of an approximate 6-foot diameter cryopanel (located on the west end chamber door (see Figure 29), complemented by two 32-inch and two 6-inch diffusion pumps in parallel backed by a single mechanical pump.

1. Bearing Test Apparatus

The test apparatus used to house and load the 100 mm bore test bearings can be seen in Figure 30 . In this view a section of the housing has been removed to expose the outer two test bearings. The center block can accommodate one or two test bearings, making four the maximum number of 100 mm bore bearings that can be tested at one time.

The arrangement for loading was similar to the setup in the small scale bearing tester. The basic difference between the two testers was that the bearings in the small scale tester were tested in a vertical position while the large scale bearing tests were conducted in a horizontal position.

2. Instrumentation

a. Loading Technique

Loading was accomplished by means of a load bolt and in the same manner as previously described in the small scale bearing tests. The load bolt or loading mechanism consisted of a 1-1/4 inch diameter steel bolt with strain gages mounted to measure the tension in pounds applied to the bolt when the nut was tightened. Strain gage measurements were made on a Baldwin SR-4 Strain Gage Indicator.

An alternate method of measuring load in the event of strain gage failure or damage was by use of a Sturtevant Torque Wrench with a 4:1 torque multiplier. The torque wrench was calibrated against strain gage measurements to read load in pounds on its own indicator.

b. System Torque

Torque measurements were accomplished by restraining the bearing housing from rotation with the load bolt acting as a torque arm, and two separate cantilever arms mounted with strain gages receiving the transmitted frictional torque of the bearing system. The load bolt seated between the instrumented cantilever arms can be seen directly under the bearing housing in

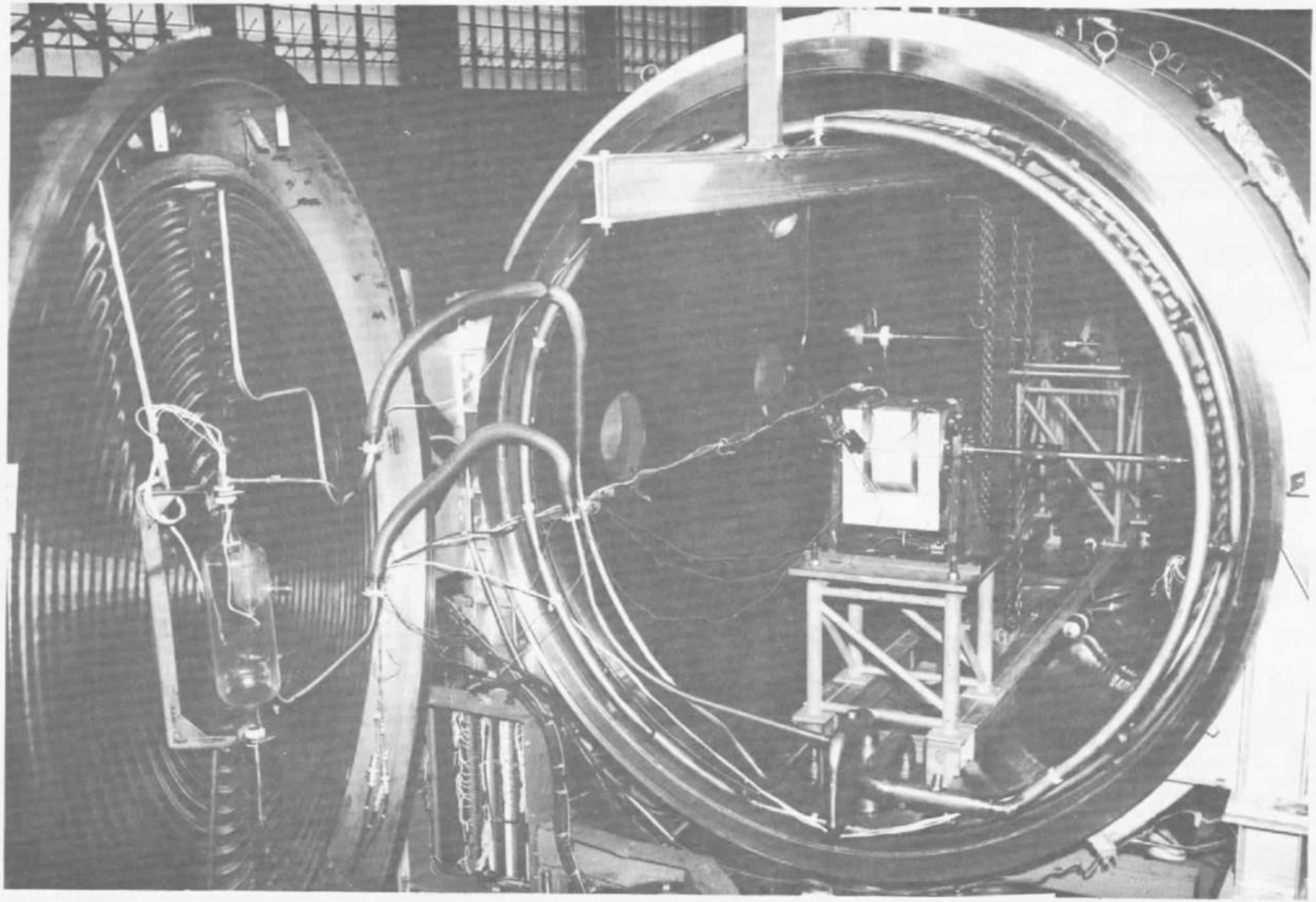

Fig. 29 7 Ft. Aerospace Chamber with Bearing Tester in Foreground and Four Square Gear Tester in Background

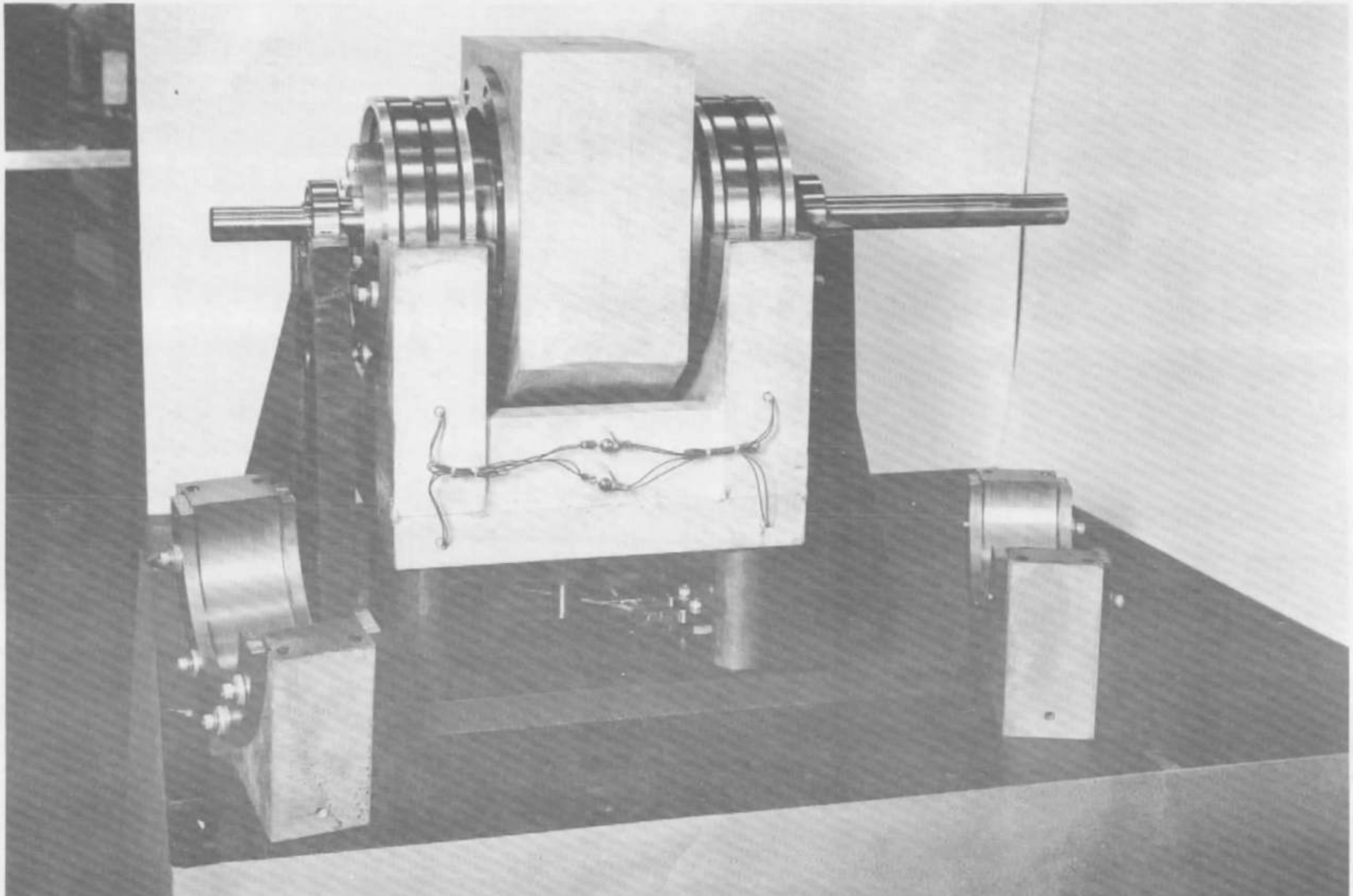

Fig. 30 Test Apparatus for Housing and Loading the 100-mm-bore Test Bearings

Figure 29 The strain gages were connected to an Ellis Bridge Amplifier whose output signal was fed to a GE Photo Electric Recorder, resulting in a continuous trace during bearing operation. The instrumented cantilever arms were calibrated to measure torque in lb-in.

c. Speed

The bearing tester motor drive was preset to rotate the bearings at 8 rpm. Each revolution of the input drive shaft actuated a microswitch which ran a cycle counter.

d. Temperature

Copper constant on thermocouples were positioned at the outer race of the test bearings. Three bearings were monitored from a temperature indicator.

e. Chamber Pressure

Chamber pressures throughout the test periods were generally maintained between 1.0×10^{-7} and 1×10^{-8} mm Hg.

Test Procedure

1. Drive motor slip clutch preset to slip at 492 lb-in.

2. Bearing (test) assembled on shaft using appropriate spacers to conform with selected bearing type configuration.

3. (a) Run-in bearings with zero load for approximate twenty-five (25) cycles in air. Record system torque.

   (b) Run-in bearings with half (1/2) load for approximate twenty-five (25) cycles in air. Record system torque.

   (c) Run-in bearings with full load applied for fifty (50) cycles in air. Record system torque.

4. After run-in of bearings with full load, the chamber was closed and vacuum pumps started.

5. Upon reaching approximately 1×10^{-7} mm Hg or better, the bearing tester was started and system torque recorded continuously.

   Tests were terminated when slippage of clutch occurred, or when the desired test duration was attained.

f. Photographs

Photographs of some of the 100 mm tested bearings are shown in Figures 31 through 37 . Some of the photographs are rather optimistic, as lubricants did wear to various degrees and torques did go up. Figures 31-34 show the condition of some of the various lubricated cylindrical bearings taken from tests where they were run in combination with spherical bearings, or were all of the same kind. Figures 35 and 36 show a spherical type bearing and Figure 37 a ball bearing.

Figure 31 shows how nicely the gold was burnished on the inner race and how well polished the rollers appeared. The races and cage only were gold plated. No breakthrough of the gold occurred. The same held true for the silver test as seen in Figure 32.

Only the races and retainer were coated with MoS_2+ graphite + silicate in Figure 33. However, one should note how uniformly the coating transferred to the rollers, resulting in excellent protection. Figure 34 shows how well the MoS_2-glass developmental coating protected the bearing elements. Note the burnished-in coating on the inner race and how polished the rollers appear. Figures 35 and 36 show different views of the same spherical bearing, having been lubricated with a film of MoS_2 + epoxy binder.

Figure 37 shows the appearance of the Apiezon L grease lubricated bearing (ball bearing inner race) after the 80-hour test. The inner race and balls were polished and in excellent condition, as was the outer race. The retainer, inadvertently reversed in the photograph, showed no pocket wear. The specially designed snap shield used in the ball bearing tests can be seen in the background with the Buna N elastomer molding beaded to it.

B. BEARING TEST RESULTS

The results of the 100 mm rolling element bearing tests are described in Table 40. The test durations shown are not all indicative of test termination due to torque increase. Some were run for a specific period and the test terminated for other reasons, such as schedule, etc. The examination results described in Table 41 should be integrated with Table 40 for a clear understanding of bearing-lubricant performance. The torque-curves (Figures 38-44) further indicate the performance characteristics of the lubricated bearings.

<u>Summary of Test Results (100 mm Bearings) and Comments</u>

1. The cylindrical type of roller bearing appeared to suffer less coating damage in general than its spherical counterpart. The simpler design of the cylindrical bearing over that of the double row spherical would appear to be a major contributing factor in better solid film performance, although the former is limited primarily to radial load usage.

2. A small to moderate material transfer can result in a substantial increase in torque when solid film lubricants are used. This is particularly true for thin metal platings. Further, test results indicated that thin platings are more sensitive than MoS_2 and graphite types of coatings to misalignment, load variations and other factors contributing to instability.

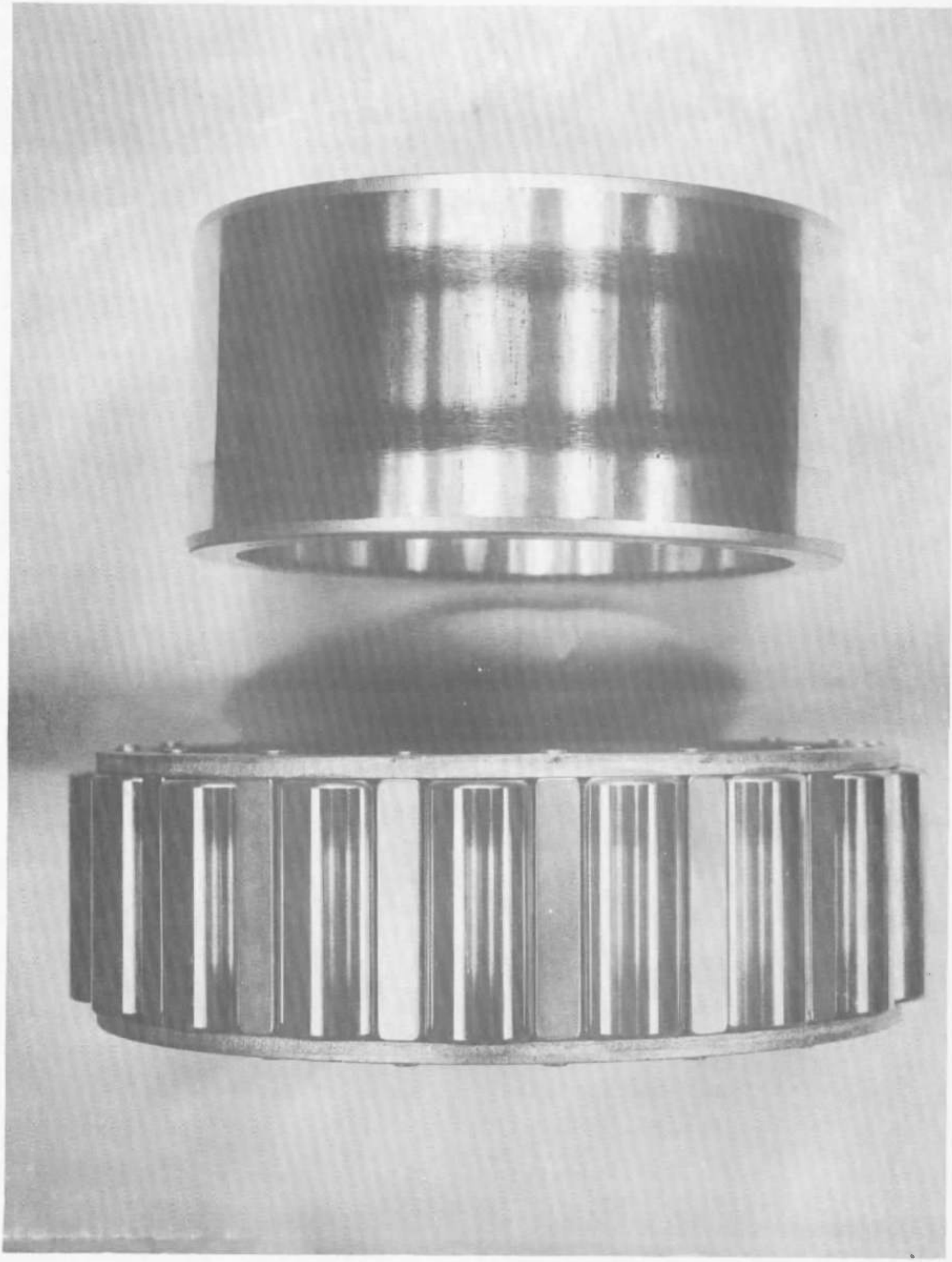

Fig. 31 Test No. 2 (one of 4 bearings) – View of (100 mm) Inner Race and Roller-cage Assembly from Cylindrical Bearing – 23 Kt Au Plated Except Rollers – Dark Bands are Polished Areas – No Coating Failure

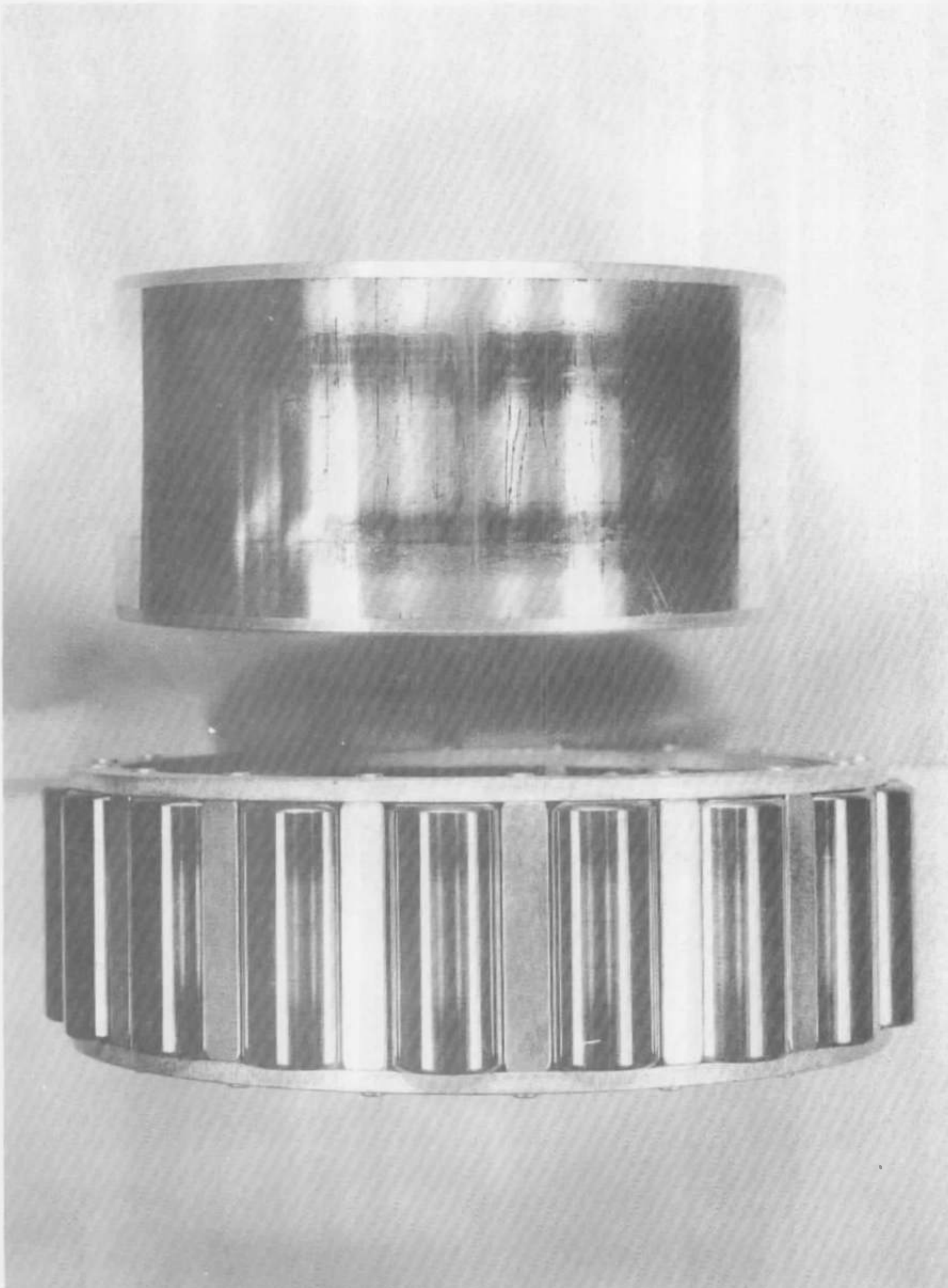

Fig. 32 Test No. 3 (one of 4 bearings) – View of Ag Plated Inner Race and Roller-Cage Assembly from Cylindrical Bearing – Coating Intact and Polished

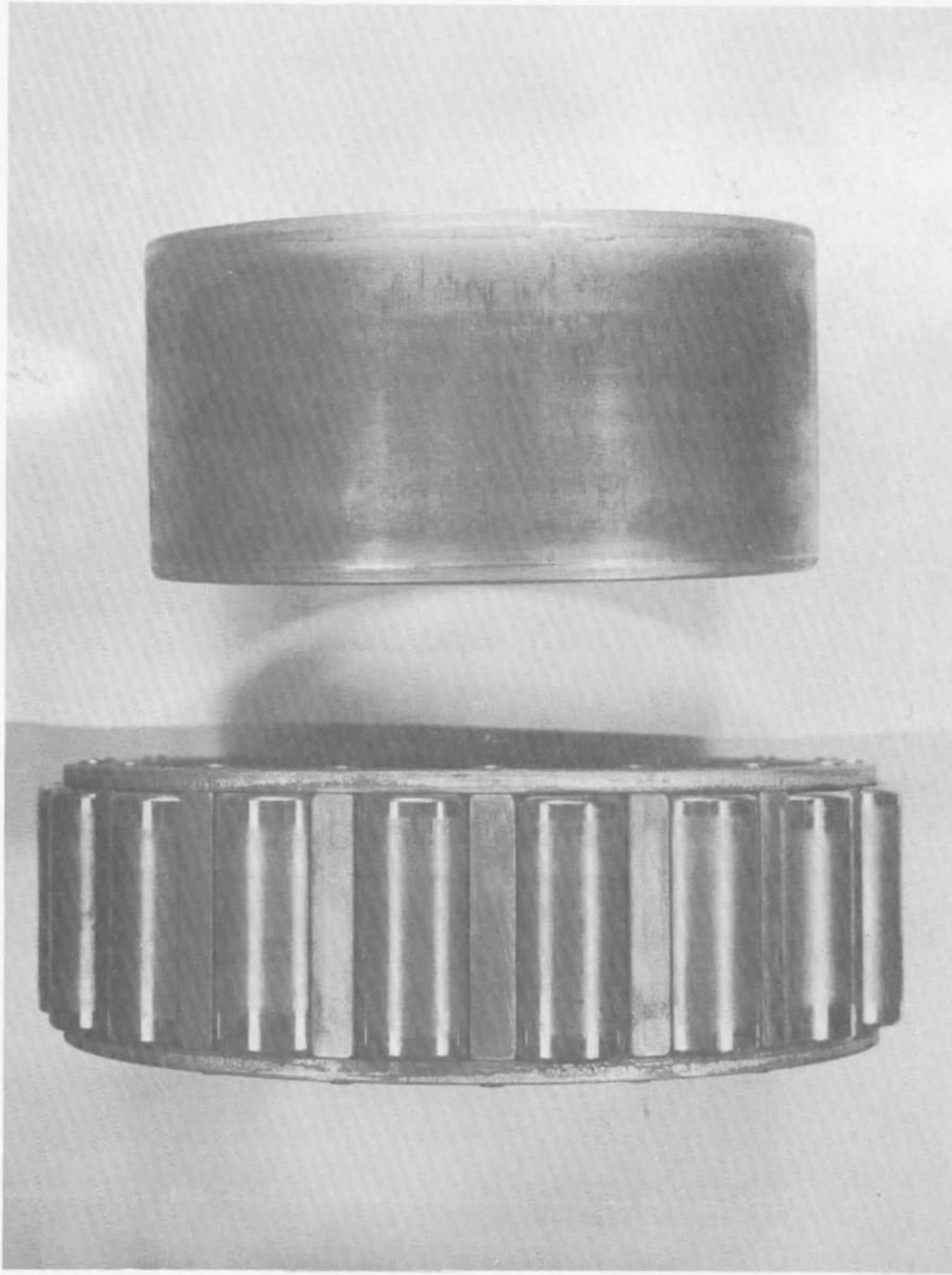

Fig. 33 Test No. 4 (one of 4 bearings) – View of I.R. and Roller-Cage Assembly from Cylindrical Bearing – Note Effectiveness of MoS_2 + Graphite + Silicate Coating on Parts – Rollers were Not Initially Coated but Now have Uniform Transfer of Coating on Surface

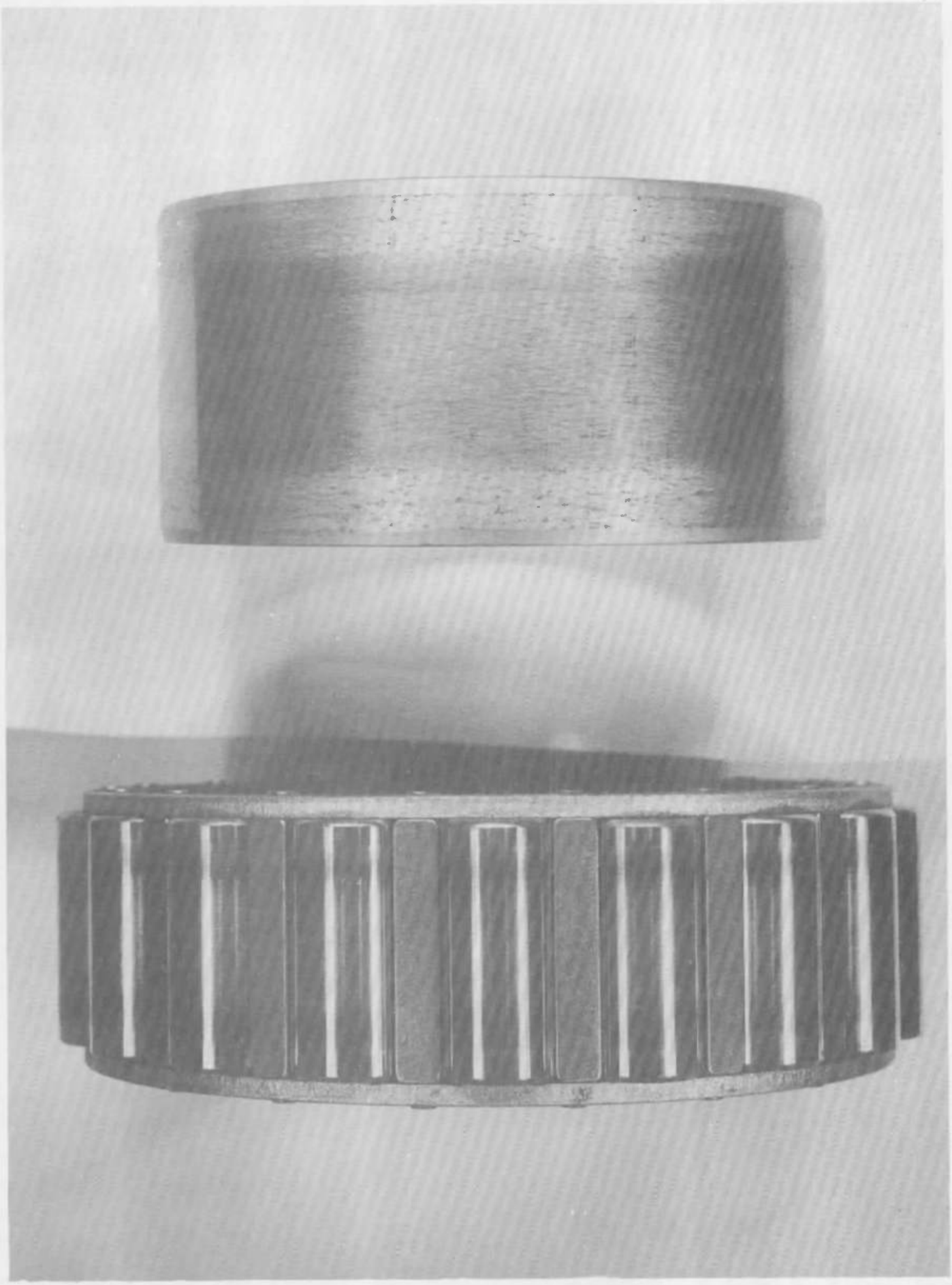

Fig. 34 Test No. 5 (one of 4 bearings) – View of I.R. and Roller-Cage Assembly from Cylindtical Bearing – ATL's Developmental Glass + MoS_2 Coating Appears Intact and "run-in"

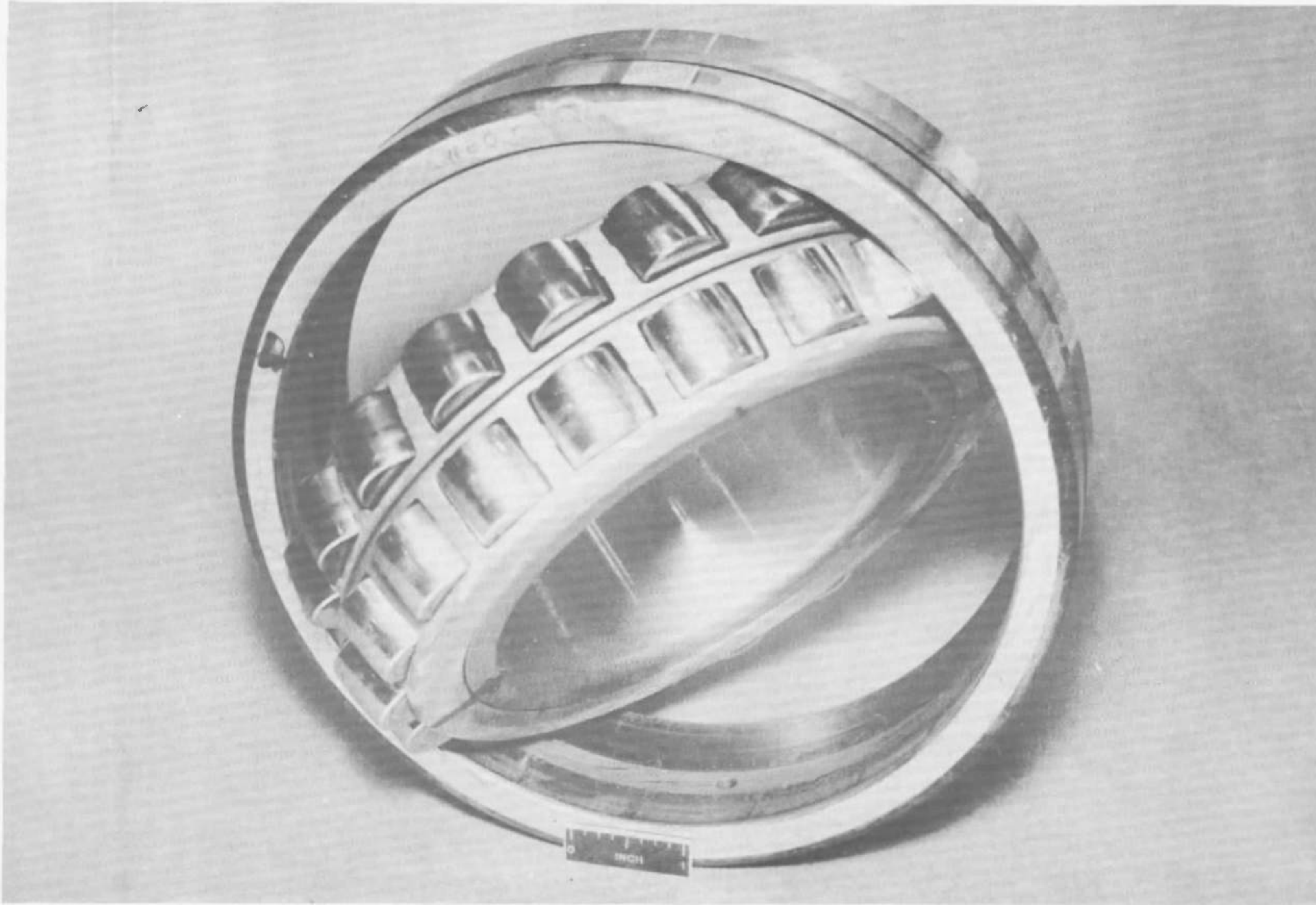

Fig. 35 Test No. 1 View of 100 mm Full Assembly Spherical Roller Bearing after Running 26.6 Hours on a Film of MoS_2 and Epoxy Binder-One Set of Rollers is Polished, The Other Set Has Picked Up a Thin Film of Lubricant

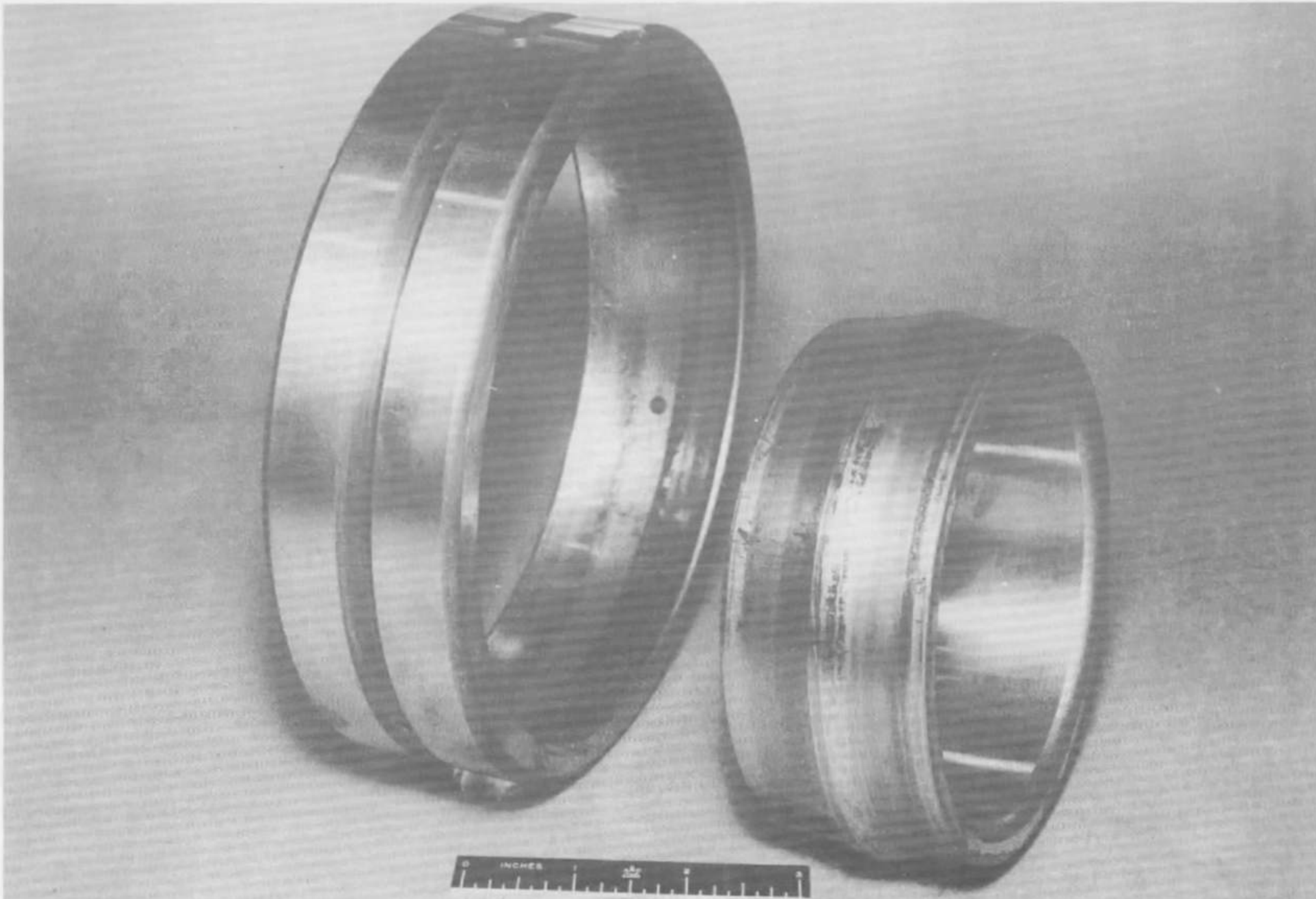

Fig. 36 Test No. 1 Spherical Bearing Shown in Fig. 7 Only with Outer and Inner Races Exposed (left to right). Although Remaining Lubricant Film was Marginal no Damage Occurred to Bearing

Fig. 37 Test No. 6 (one of 4 bearings) – View of I.R., Balls, Special Snap-in Shield, and Section of Retainer from 100 mm Ball Bearing – Test Results were Excellent as Denoted by Highly Polished Surfaces – (Lub – Apiezon L Petroleum Distillate)

TABLE 40

ATL LARGE SCALE BEARING (100 MM) TEST RESULTS

TEST NO.	BEARING TYPE (100 MM)	NO. OF BRGS.	RADIAL LOAD (lb/brg) P_1	P_2	BEARING STATIC LOAD CAPACITY (Co) lbs.	Co/P[1] RATIO Co/P_1	Co/P_2	LUBRICANT[2]	SPEED (RPM)	TOTAL CYCLES	TEST DURATION IN VAC. (HRS.)
	Bearings										
1	SKF 2220CY Spherical EB 5220B Cylindrical	2 2	5,600 5,600		56,000 71,000	10.0 12.7		MoS_2 + Epoxy Binder	10	14,101	26.6
2	SKF 2220CY Spherical EB 5220B Cylindrical	2 2	4,000 4,000		56,000 71,000	14.0 17.7		23 Kt Gold	8	1,938	4.4
3	EB 5220B Cylindrical	4	5,100		71,000	14.0		Silver	8	6,239	12.5
4	EB 5220B Cylindrical	4	5,100	7,100	71,000	14.0	10.0	MoS_2 + Graphite + Silicate Binder	8	62,640 + 46,859 109,499	130.0 + 100.0 230.0
5	SKF 2220CY Spherical EB 5220B Cylindrical	2 2	4,000 4,000		56,000 71,000	14.0 17.7		MoS_2 + Glass	8	24,765	53.3
6	220A Ball Bearings	4	2,850		28,500	10.0		Low vapor[3] petroleum distillate grease (Apiezon L)	8	36,302	80.0
7	SKF 2220CY Spherical EB 5220B Cylindrical	2 2	4,000 4,000		56,000 71,000	14.0 17.7		MoS_2 + Epoxy Binder	8	45,205	95.8
8	SKF 2220CY Spherical EB 5220B Cylindrical	2 2	4,000 4,000		56,000 71,000	14.0 17.7		23 Kt Gold	8	217	0.5

(For explanation of footnotes, see next sheet.)

TABLE 40 (Continued)

Footnotes:

1. C_o/P is the ratio of the static load capacity of the bearing to the applied load where C_o is defined as that radial load which will result in a permanent deformation of 0.0001 times the diameter of the rolling element. P is the applied radial load to the bearing.

2. Where solid film type of lubrication was used, all bearing surfaces were coated, with the exception of the rollers. Where the low vapor pressure greases were used, all bearing surfaces were coated including the rollers.

3. Scheduled for 80 hour test only - excellent performance - no failure. Apiezon L is a low vapor pressure petroleum oil distillate grease with a vapor pressure reported as 10^{-10} - 10^{-11} mm of Hg at room temperature.

4. Tests were conducted in vacuum between 1×10^{-7} and 1×10^{-8} mm of Hg.

TABLE 41

EXAMINATION RESULTS OF ATL LARGE SCALE BEARING TESTS (100 MM)

CYLINDRICAL ROLLER BEARING ROLLWAY EB5220B	SPHERICAL ROLLER BEARING SKF2220CY	SPHERICAL ROLLER BEARING SKF2220CY	CYLINDRICAL ROLLER BEARING ROLLWAY EB5220B
Test No. 1 Lub - MoS_2 + Epoxy Binder Envir - Vac (5.2×10^{-8} mm Hg Max)			
ROLLERS-Thin film of lubricant deposited on surface of rollers - slight buildup of lubricant near ends of some rollers.	ROLLERS-Dull gray color over adjacent rollers - no damage. Polished with some lub. buildup over drive half.	ROLLERS-Thin film of lubricant on rollers on drive side - adjacent side rollers polished with some coating pickup.	ROLLERS-Thin film of lubricant deposited on rollers.
OR-Coating intact over major portion of outer race - surface coating thin in load zone area but still effective.	OR-Coating marginal over most of roller contact area.	OR-Coating marginal over drive side - coating intact over adjacent side.	OR-Coating intact over major portion of outer race surface - coating thin in load zone area but still effective.
IR-Initial coating thickness reduced considerably over roller contact area, however a remaining effective thin film was observed.	IR-Coating polished on drive side - lub. thin but effective - adjacent side dull with coating marginal.	IR-Coating polished and intact on adjacent side-drive side coating marginal in some areas.	IR-Initial coating thickness reduced considerably, however a remaining thin film was observed.
(*)	CAGES-Light polishing of lubricant on drive side. Coating polished and intact over most of opposite cage. Small areas of brass exposure in axial direction.	CAGES-Moderate wear over outermost contact point, coating intact over rest of cage (drive side). Coating in excellent condition on adjacent cage.	CAGE-Coating on cage segments intact and in excellent condition. No wear observed.
SR-Coating polished in some areas - coating intact.	SSR-Excellent condition.	SSR-Excellent condition.	SR-Coating polished in some areas - coating intact.
Bearing turned freely.	Bearing turned with some difficulty.	Bearing jamming.	Bearing turned freely.
BRG. NO. 9-C-9	BRG. NO. 7-S-7	BRG. NO. 8-S-8	BRG. NO. 10-C-10

Comments: It would appear that a combination of marginal lubrication, buildup of coating on parts, and cage wear was responsible for 8-S-8 jamming and the subsequent termination of the test.

(*) Cage-roller assembly left intact for observational purposes; therefore unable to examine contact areas of cage segments.

TABLE 41 (Continued)

CYLINDRICAL ROLLER BEARING EB5220B	SPHERICAL ROLLER BEARING SKF2220CY	SPHERICAL ROLLER BEARING SKF2220CY	CYLINDRICAL ROLLER BEARING SKF2220CY
23 kt Test No. 2 Lub - Gold Plating Envir - Vac(1.4 x 10^{-7} MM Hg Max.)			
ROLLERS-Surfaces in excellent condition - rollers polished.	ROLLERS-Highly polished - few light score marks(drive) - highly polished on aft side.	ROLLERS-Highly polished - single score marks on several of the rollers(aft) - roller surfaces excellent on drive side.	ROLLERS-Surfaces in excellent condition - rollers polished.
OR-Excellent condition - coating intact and lightly polished.	OR-Coating intact - contact area highly polished. Light flaking on aft side - light buildup on outer periphery of drive side.	OR-Coating intact - contact area highly polished - some gold particles built up on load zone area with evidence of some light flaking.	OR-Excellent condition - coating intact and lightly polished.
IR-Same as OR.	IR-Coating intact - light flaking and buildup in load zone (aft and drive respectively).	IR-Coating intact - both sides polished - light pickup on aft side.	IR-Same as OR.
CAGE-Coating intact on cage segments. Excellent condition.	CAGE-Coating intact - light polishing on both cages.	CAGES-Coating polished on high contact points. Possibly light wear-(aft). Coating intact - no wear on drive side.	(*)
SR-Excellent condition -	SR-Coating intact - excellent condition.	SR-Coating intact - excellent condition.	SR-Excellent condition -
Bearing turned freely. BRG. NO. 6-C-6	Bearing turned freely. BRG. NO. 2-S-2	Bearing turned freely. BRG. NO. 1-S-1	Bearing turned freely. BRG. NO. 5-C-5

Comments: Small buildup of Au between ID of cage and race resulted in torque increase. No actual plating failure occurred over any of the four bearings.

(*) Cage-roller assembly left intact for observational purposes; therefore unable to examine contact areas of cage segments.

TABLE 41 (Continued)

EXAMINATION RESULTS OF LARGE SCALE BEARING TESTS (100 MM)

CYLINDRICAL ROLLER BEARING EB5220B	CYLINDRICAL ROLLER BEARING EB5220B	CYLINDRICAL ROLLER BEARING EB5220B	CYLINDRICAL ROLLER BEARING EB5220B
Test No. 3 Lub-Silver Plate, Envir - Vac (6 x 10^{-8} mm Hg Max.)			
ROLLERS-Polished surfaces - rollers in good condition.	ROLLERS-Polished surfaces - rollers in good condition.	ROLLERS-Polished surfaces - rollers in good condition.	ROLLERS-Polished surfaces - rollers in good condition.
OR-Coating intact - load zone considerably polished.	OR- Major portion of coating intact. Area in load zone however indicates some peeling of silver although not to copper strike on base metal. Light buildup in few spots.	OR-Coating intact - load zone considerably polished. Light scuffing in some areas.	OR-Coating intact - load zone considerably polished.
IR-Coating intact and polished - excellent condition.	IR-Coating polished and intact - light pickup on one side.	IR-Major portion of coating intact however a circumferential band approx. 1/4" wide, "chatter marked" in appearance, appears depleted of lubricant (drive end side).	IR-Major portion of coating intact - several small bands within 90 degree sector indicated light spalling of the silver had occurred, revealing the copper strike - no damage to base metal evidenced.
CAGE-Coating in excellent condition on cage segments.	CAGE-Cage segments in excellent condition - coating intact and polished where rollers contact.	(*)	(*)
SR-Polished.	SR-Polished groove on snap ring - coating generally intact. Possible light wear.	SR-Polished groove on snap ring surface - major portion of coating intact.	SR-Polished groove - major portion of coating intact.
Bearing turned freely.	Bearing turned freely.	Bearing turned freely.	Bearing turned freely.
BRG. NO. 4-C-4	BRG. NO. 3-C-3	BRG. NO. 2-C-2	BRG. NO. 1-C-1

Comments: It would appear that the thrust load and moderate wear on the snap ring surfaces resulted in high enough torque to cause slippage of the motor drive slip clutch.

(*) Cage-roller assembly left intact for observational purposes; therefore unable to examine contact areas of cage segments.

TABLE 41 (Continued)

CYLINDRICAL ROLLER BEARING EB5220B	CYLINDRICAL ROLLER BEARING EB5220B	CYLINDRICAL ROLLER BEARING EB5220B	CYLINDRICAL ROLLER BEARING EB5220B
	Test No. 4 Lub - MoS_2 + Graphite + Silicate Binder Envir Vac (3.3×10^{-8} mm Hg max.)		
ROLLERS-Surfaces consist of polished and burnished circumferential bands - some evidence of lubricant pickup.	ROLLERS-Light film of lubricant observed on roller surfaces.	ROLLERS-Light film of lubricant observed on rollers.	ROLLERS-Surfaces consist of polished and burnished circumferential bands - little evidence of coating pickup.
OR-Coating intact over major portion of outer race surface - load zone area highly polished. Metallic luster indicating depletion of coating. No damage.	OR-Coating intact over major portion of outer race surface - coating marginal over load zone area.	OR-Coating intact over major portion of outer race surface - coating marginal over load zone area.	OR-Coating intact over major portion of outer race surface - highly polished metallic luster over load zone.
IR-Lubrication marginal over considerable portion of contact area - metallic areas polished - no damage.	IR-Coating marginal over contact area.	IR-Coating marginal over contact area - some coating pickup.	IR-Highly polished contact area - metallic appearance indicative of coating depletion.
CAGE-Coating intact on cage segments - excellent condition - additional coating buildup on small portion of some segments.	CAGE-Coating intact on cage segments - excellent condition - light coating buildup at ends of segments - no damage.	CAGE- Coating intact - excellent condition - slight coating buildup at ends of the cage segments.	(*)
SR-Coating intact - polished in some areas.	SR-Coating intact - polished in some areas.	SR-Coating intact - polished in some areas.	SR-Coating intact - polished in some areas.
Bearing turned freely.	Bearing turns freely.	Bearing turned freely.	Bearing turns freely.
BRG. NO. 13-C-13	BRG. NO. 15-C-15	BRG. NO. 14-C-14	BRG. NO. 16-C-16

Comments: This test was successful over the period tested and stopped only for schedule purposes. The lubricant was subjected to two load conditions. The test could have continued even though lubrication was becoming marginal in some areas.

(*) Cage-roller assembly left intact for observational purposes; therefore unable to examine contact areas of cage segments.

TABLE 41 (Continued)

CYLINDRICAL ROLLER BEARING ROLLWAY EB5220B	SPHERICAL ROLLER BEARING SKF2220CY	SPHERICAL ROLLER BEARING SKF2220CY	CYLINDRICAL ROLLER BEARING ROLLWAY EB5220B
	Test No. 5 Lub. MoS_2 + ATL Glass Binder Envir. Vac (2.0×10^{-8} mm Hg Max.)		
(*)	ROLLERS-Highly polished in one row, light discoloration band - no apparent transfer of coating. Second row smooth, polished, discolored - thin glass film observed.	ROLLERS-Highly polished in one row, light discoloration bands, no damage. Second row - smooth, polished, discolored - discoloration due to thin glass film and film of bronze from cage wear - no damage to roller material.	ROLLERS-Polished glassy appearance - light gray over roller surface - extremely thin protective film observed - excellent condition.
	OR-Coating appears intact except one area located opposite IR portion which also was depleted of coating. No apparent damage to OR surface.	OR-60% of coating intact area indicating depletion, also shows film of bronze indicating plastically deformed wear debris from cage.	OR-Thin film of coating remaining - some buildup of coating platelets - no evidence of damage - not as much coating remaining on OD as ID.
	IR-Coating partially intact on side with highly polished rollers. Coating appears depleted on opposite side. No damage to IR surface noted.	IR-Coating effective on one race surface. On opposite side shiny thin film of bronze observed.	IR-Coating intact - excellent condition.
	CAGES-Coating intact - excellent condition on one cage - same for other except for some light pocket wear on one face of four contact faces.	CAGES-Coating intact - excellent condition for one cage. Other cage mild wear on two contact faces.	CAGE-Coating on cage segments appears to be excellent - polished glassy appearance.
	SR-Coating intact - excellent condition.	SR-Coating intact, excellent condition.	SR-Coating polished and intact.
BRG. NO. 19-C-19	BRG. NO. 11-S-11	BRG. NO. 12-S-12	BRG. NO. 20-C-20

Comments: An improved method of applying this developmental type coating on bearing and gear surfaces may improve performance over the present method of application.

(*) Bearing assembly left intact for observational purposes.

TABLE 41 (Continued)

BALL BEARING NORMA HOFFMANN 6220-ZZ (220PP)	BALL BEARING NORMA HOFFMANN 6220-ZZ (220PP)	BALL BEARING NORMA HOFFMANN 6220-ZZ (220PP)	BALL BEARING NORMA HOFFMANN 6220-ZZ (220PP)
	Test No. 6 Lub Apiezon L, Petroleum Distillate Grease Envir Vac(1.3 x 10^{-8} mm Hg Max.)		
BALLS-Highly polished, excellent condition.	BALLS-Highly polished, excellent condition.	(*)	(*)
IR-Highly polished, excellent condition.	IR-Highly polished - excellent condition.		
OR-Highly polished, excellent condition.	OR-Highly polished, excellent condition.		
CAGE-Excellent condition - polished in several contact areas - no evidence of wear.	CAGE-Cage ball pockets in excellent condition - no indication of wear.		
SNAP SHIELDS*-Excellent condition - no evidence of any effects from lubricant.	SNAP SHIELDS-Excellent condition - no evidence of any effects of lubricant on elastomer.		
BRG. NO. 1-B-1	BRG. NO. 2-B-2	BRG. NO. 3-B-3	BRG. NO. 4-B-4

Comments: For comparison purposes, Apiezon L was tried as a bearing grease lubricant, for a period of 80 hours. The results were excellent as indicated by the examination.

* Special snap rings were designed for the purposes of minimizing lubricant creepage, and allowing the bearing to be lubricated periodically quite easily. A shield mold was made in which Buna N was molded on a AISI 1010 steel ring, the two components comprising the bearing shield. This elastomeric attachment allowed the bearing shield to be snapped into place or removed quite easily and at the same time provided an excellent seal against loss of lubricant. The shields were made specifically for the Norma Hoffmann 6220-22 (220 PP) ball bearing tests, however, it is expected that this method of shielding can be used on other types and sizes of bearings.

(*) Bearing assembly left intact for observational purposes. Bearings turn freely.

TABLE 41 (Continued)

CYLINDRICAL ROLLER BEARING ROLLWAY EB5220B	SPHERICAL ROLLER BEARING SKF2220CY	SPHERICAL ROLLER BEARING SKF2220CY	CYLINDRICAL ROLLER BEARING ROLLWAY EB5220B
	Test No. 7 Lub: MoS_2 + Epoxy Binder Envir - Vac (3.9×10^{-8} mm Hg Max.)		
ROLLERS-Major portion of roller contact area coated with thin film (gray) of effective MoS_2 coating.	ROLLERS-Thin dark gray and bronze color coating on one set of rollers - other set of rollers show band of MoS_2 coating only. No apparent damage to roller.	ROLLERS-Major portion of roller contact coated with thin effective coating of MoS_2 - second set of rollers polished with smaller portion of coating on roller surface - good condition.	ROLLERS-Major portion of roller contact area coated with thin gray film of effective MoS_2 coating - rollers appear in good condition.
IR-Coating worn but thin film still intact and apparently still effective.	IR-Half of race surface effectively coated with MoS_2 - other half coated with MoS_2 and brass, surprisingly, uniformly.	IR-Major portion of IR effectively coated - some areas highly polished - some areas coating marginal.	IR-Coating appears considerably worn from standpoint of original thickness - remainder of coating, while thin, appears to still be effective.
OR-Variation of coating thickness over race surface - major portion of race coated to some degree - small area in load zone, appears marginal.	OR-Coating effective over approx. 60% of surface - coating marginal, or of brass composition over remainder of surface.	OR-Same as IR.	OR-Variation of sufficiently thick coating in some areas to marginal in small portion of load zone.
CAGE- (*)	CAGES-One cage in excellent condition - other cage incurred some brass wear.	CAGES-Coating intact, wear nil - good condition.	CAGE- Coating on cage segments in tact and in excellent condition.
SR-General condition good - approx. 3" band on one side polished and coating worn.	SR-Coating intact - good condition.	SR-Coating appears intact.	SR-General condition good - some polishing.
BRG. NO. 11-C-11	BRG. NO. 5-S-5	BRG. NO. 6-S-6	BRG. NO. 12-C-12

Comments: Cage wear on Brg. 5-S-5 and subsequent buildup of wear debris appeared to be major factor associated with torque increase. Marginal lubrication in some areas was also a contributor.

(*) Cage and roller assembly intact for observational purposes.

TABLE 41 (Continued)

CYLINDRICAL ROLLER BEARING ROLLWAY EB5220B	SPHERICAL ROLLER BEARING SKF2220CY	SPHERICAL ROLLER BEARING SKF2220CY	CYLINDRICAL ROLLER BEARING ROLLWAY EB5220B
	Test No. 8 Lub: 23 kt Gold Plate, Envir Vac(4 x 10^{-8} mm Hg Max.)		
ROLLERS-Highly polished - excellent condition.	ROLLERS-Polished, light pickup of wear debris (av and bronze) on some rollers. Second row of roller polished. No pickup - excellent condition.	ROLLERS-Polished and in excellent condition - no pickup.	ROLLERS-Highly polished - excellent condition.
IR-Coating intact - polished and in excellent condition.	IR-Coating effective on race surface - coating burnished, polished and intact - some pickup on outer periphery and where ID of spacer ring - contacts inner race.	IR-Coating intact - polished and in excellent condition.	IR-Coating intact - polished and in excellent condition.
OR-Coating intact - polished and excellent condition.	OR-Coating intact, some coating wear - polished - light pickup.	OR-Coating intact - polished, excellent condition.	OR-Coating intact - polished especially in load zone - excellent condition.
CAGE-Cage segment coating intact and in excellent condition.	CAGES-One cage in excellent condition - other cage burnishing and light wear.	CAGES-Coating lightly buffed - wear nil. Excellent condition.	CAGE-Cage appears to be in excellent condition.
SR- Excellent condition - untouched.	SR-General condition excellent except for small deep groove 1/8" long on one face. Could have been made prior to assembly of brg. or upon dis-assembly.	SR-General condition good - some buffing on ID of spacer.	SR-Excellent condition - little contact evidenced.
BRG. NO. 7-C-7	BRG. NO. 3-S-3	BRG. NO. 4-S-4	BRG. NO. 8-C-8

Comments: General performance of Au plate was good as attested to by three of the four tested bearings - fourth (3-S-3) appears to have been responsible for torque increase.

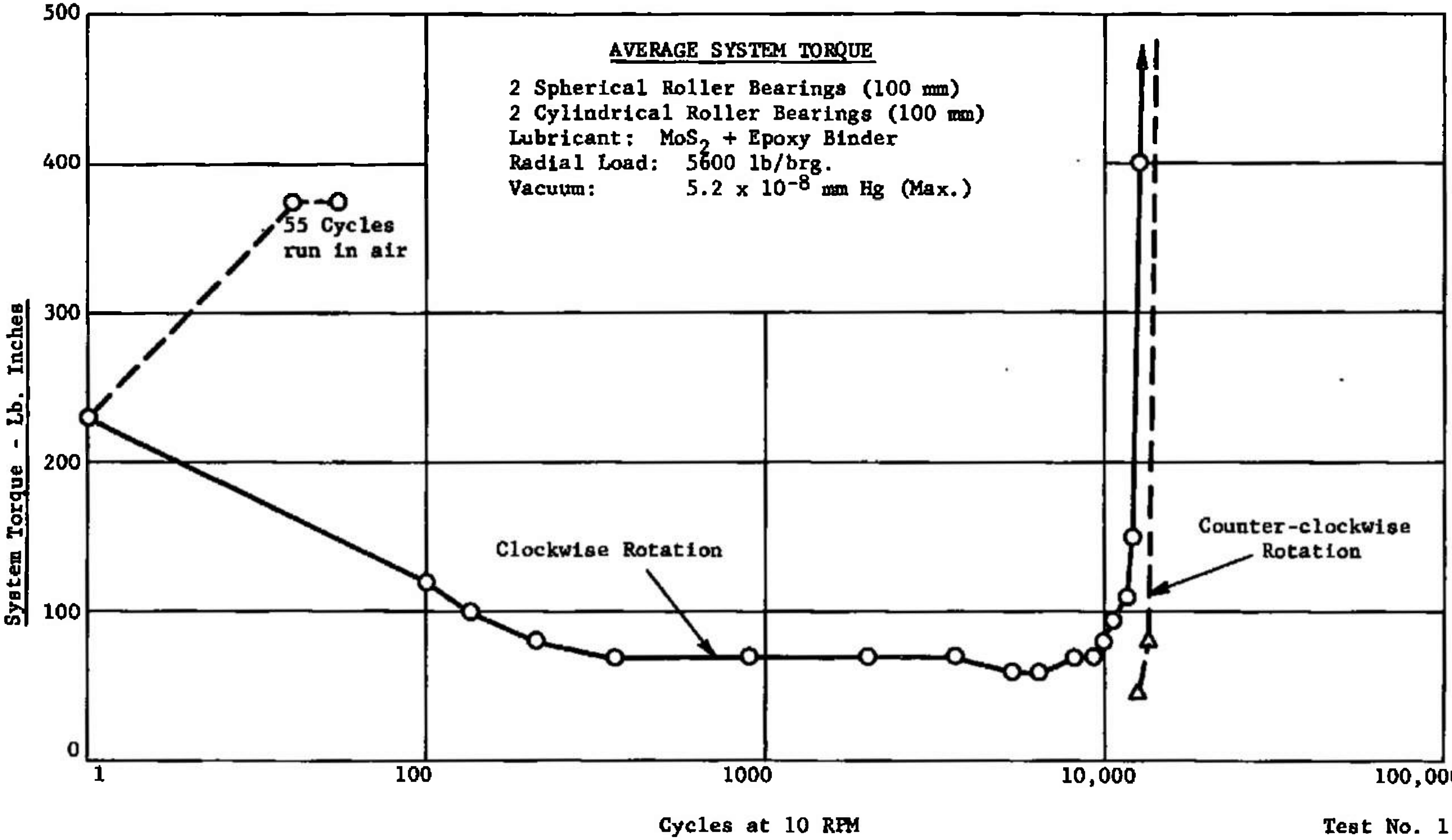

Fig. 38 Average System Torque Curves

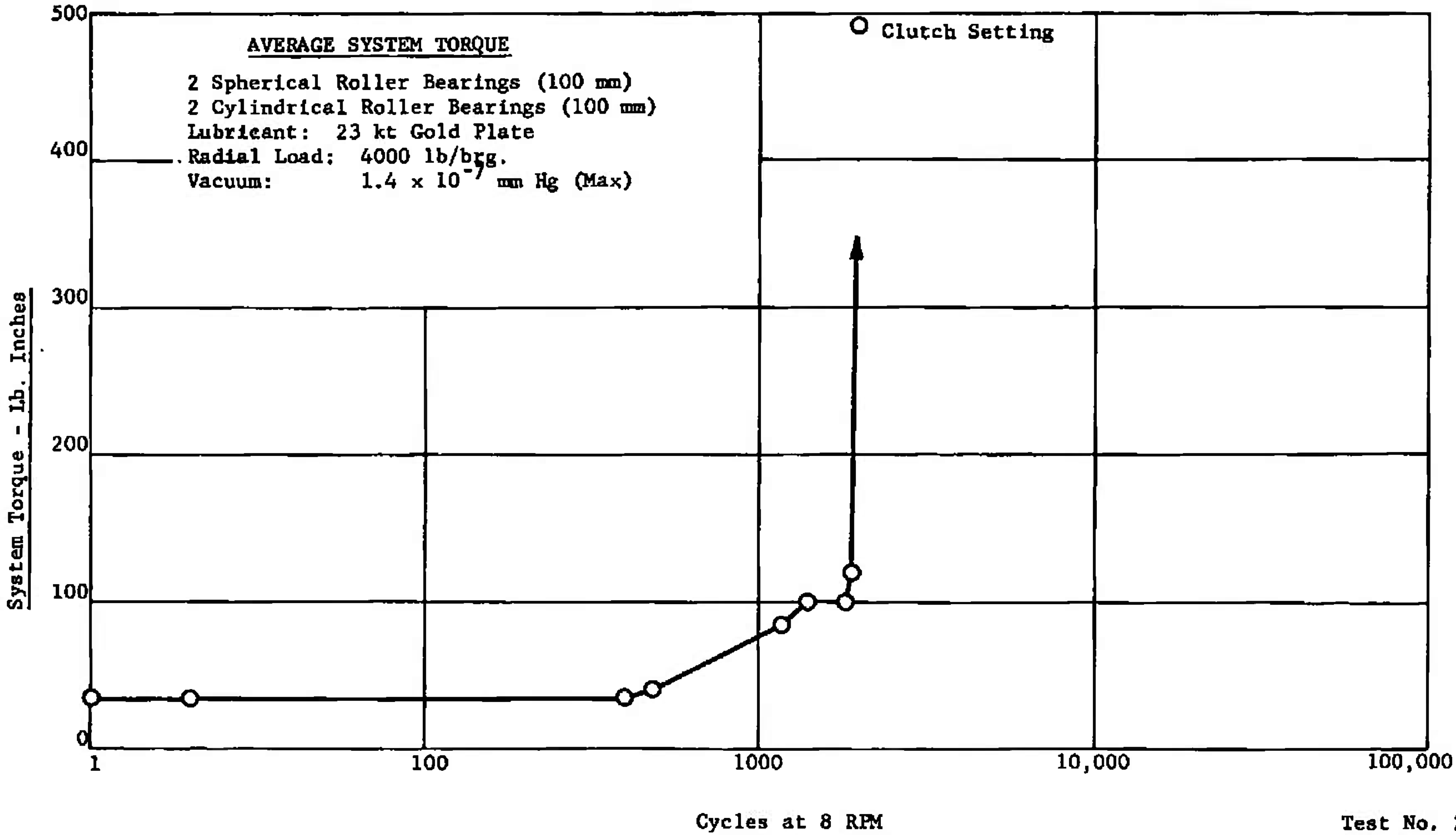

Fig. 39 Average System Torque Curves

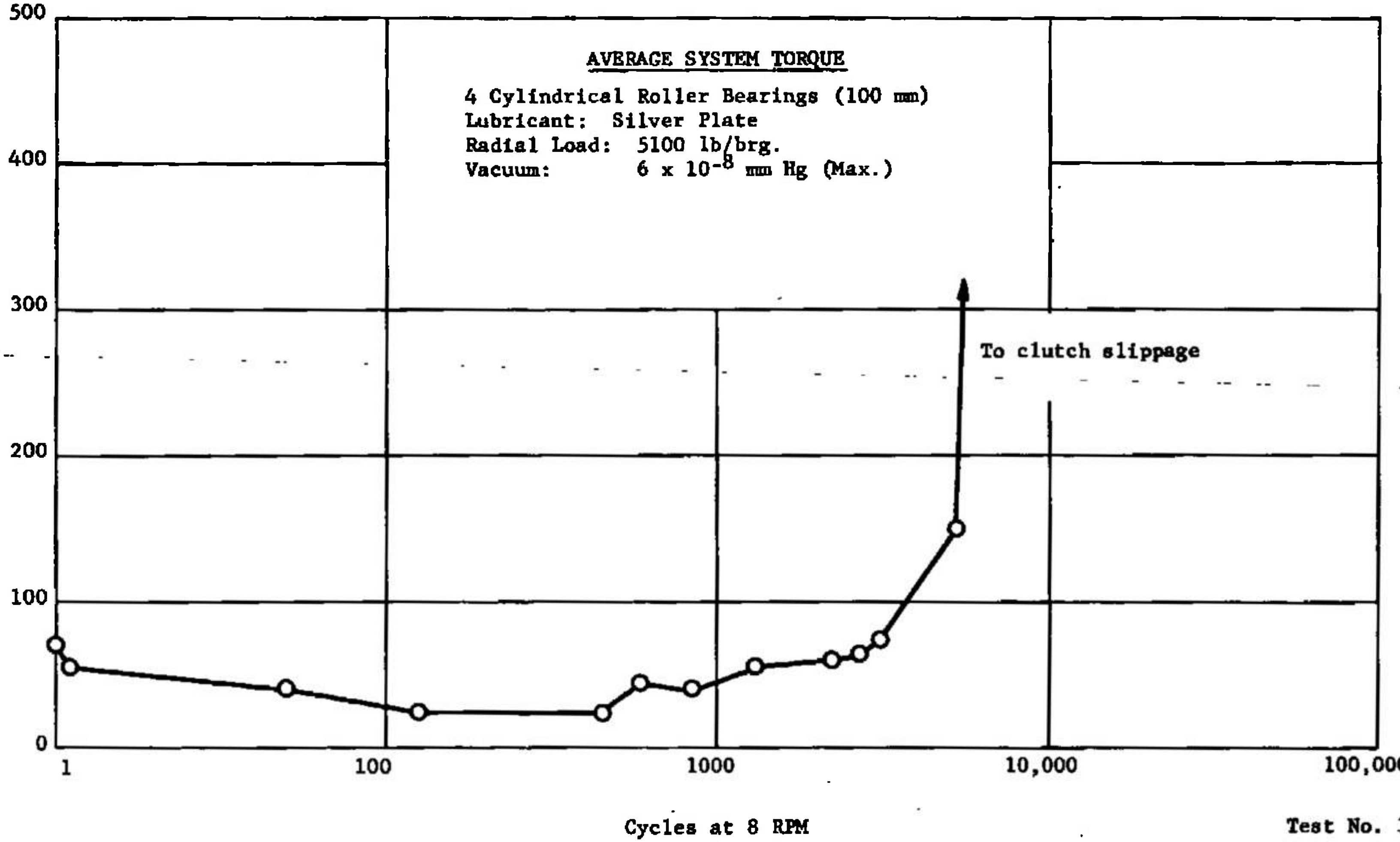

Fig. 40 Average System Torque Curves

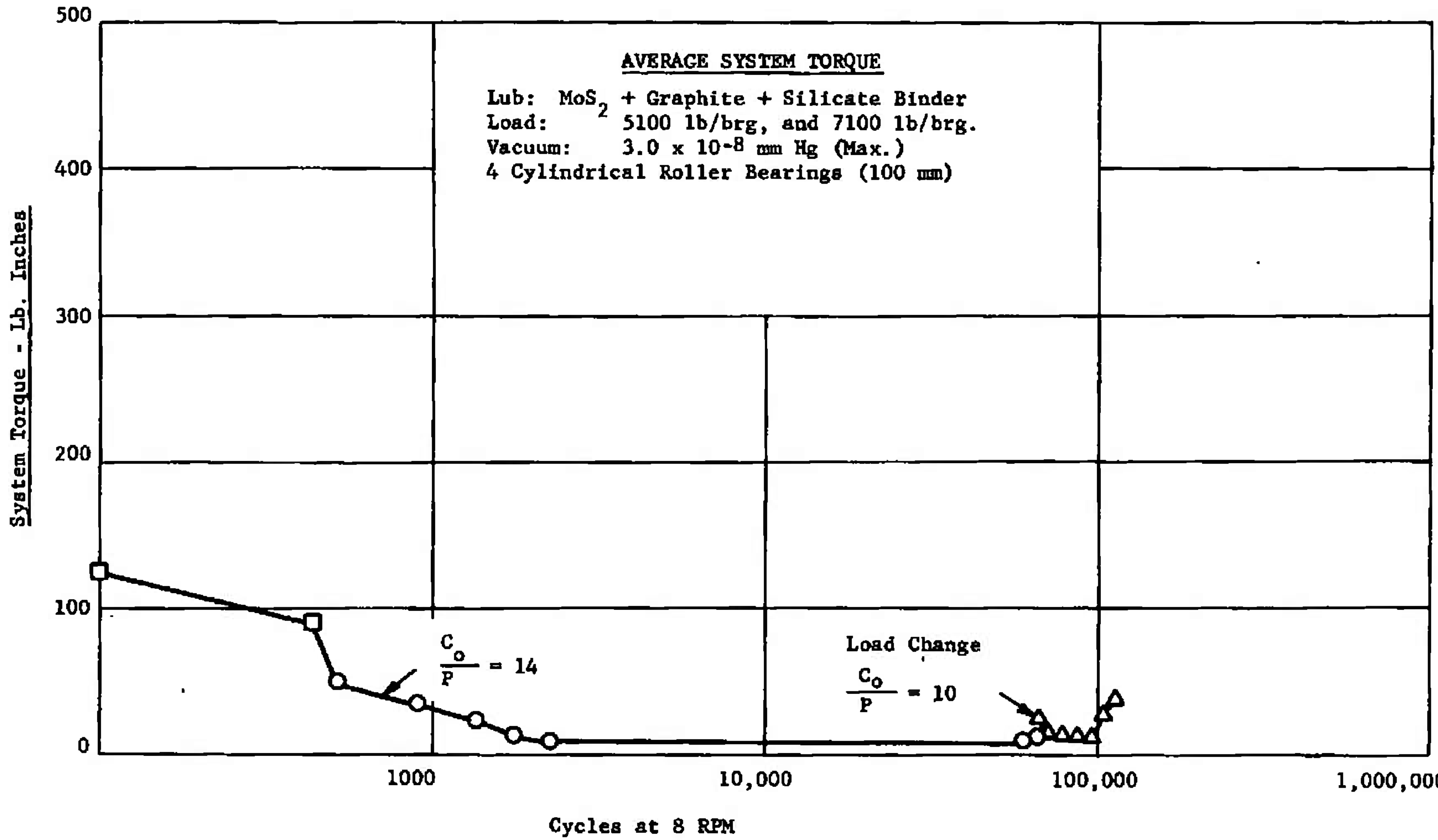

Fig. 41 Average System Torque Curves

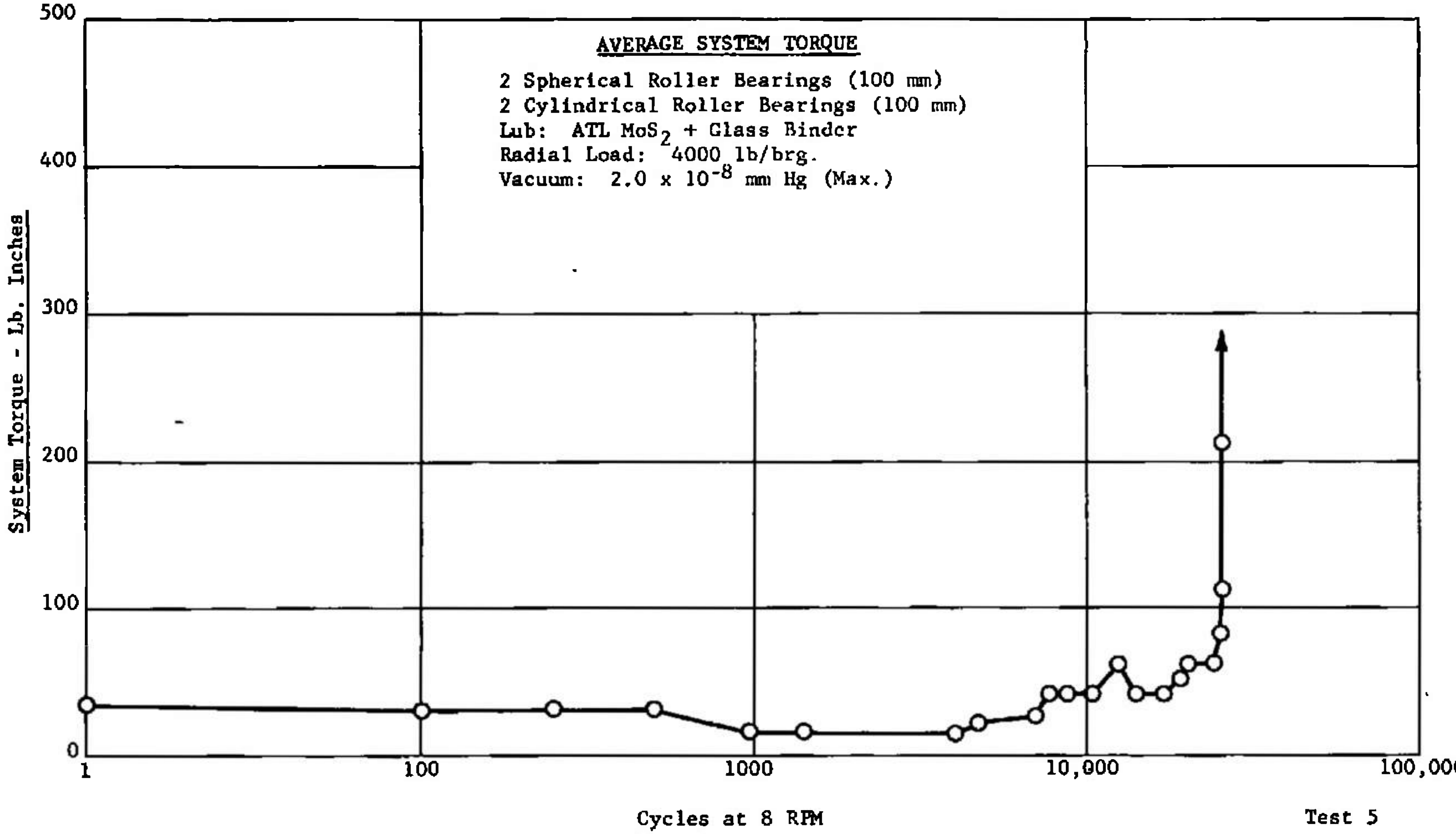

Fig. 42 Average System Torque Curves

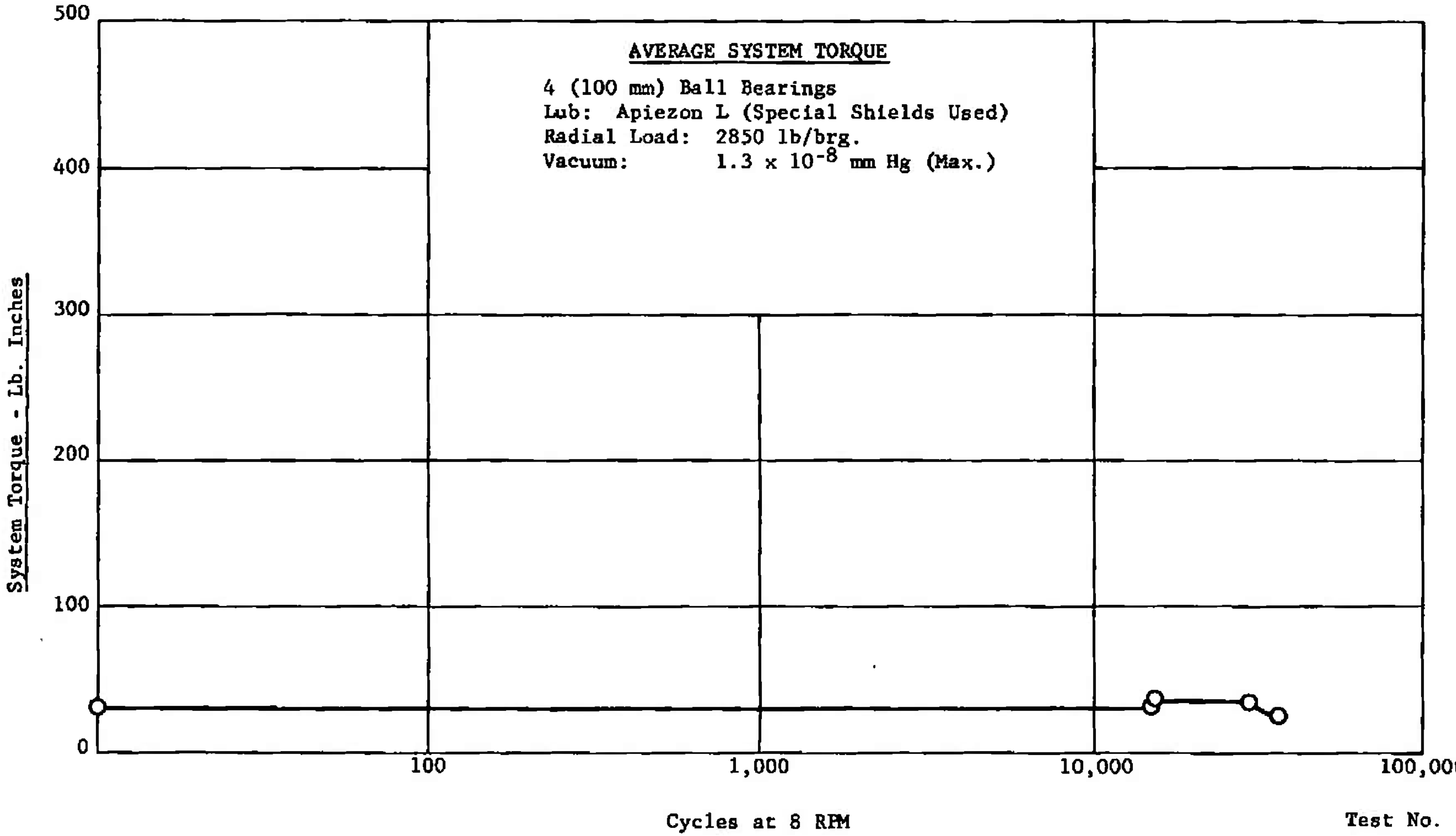

Fig. 43 Average System Torque Curves

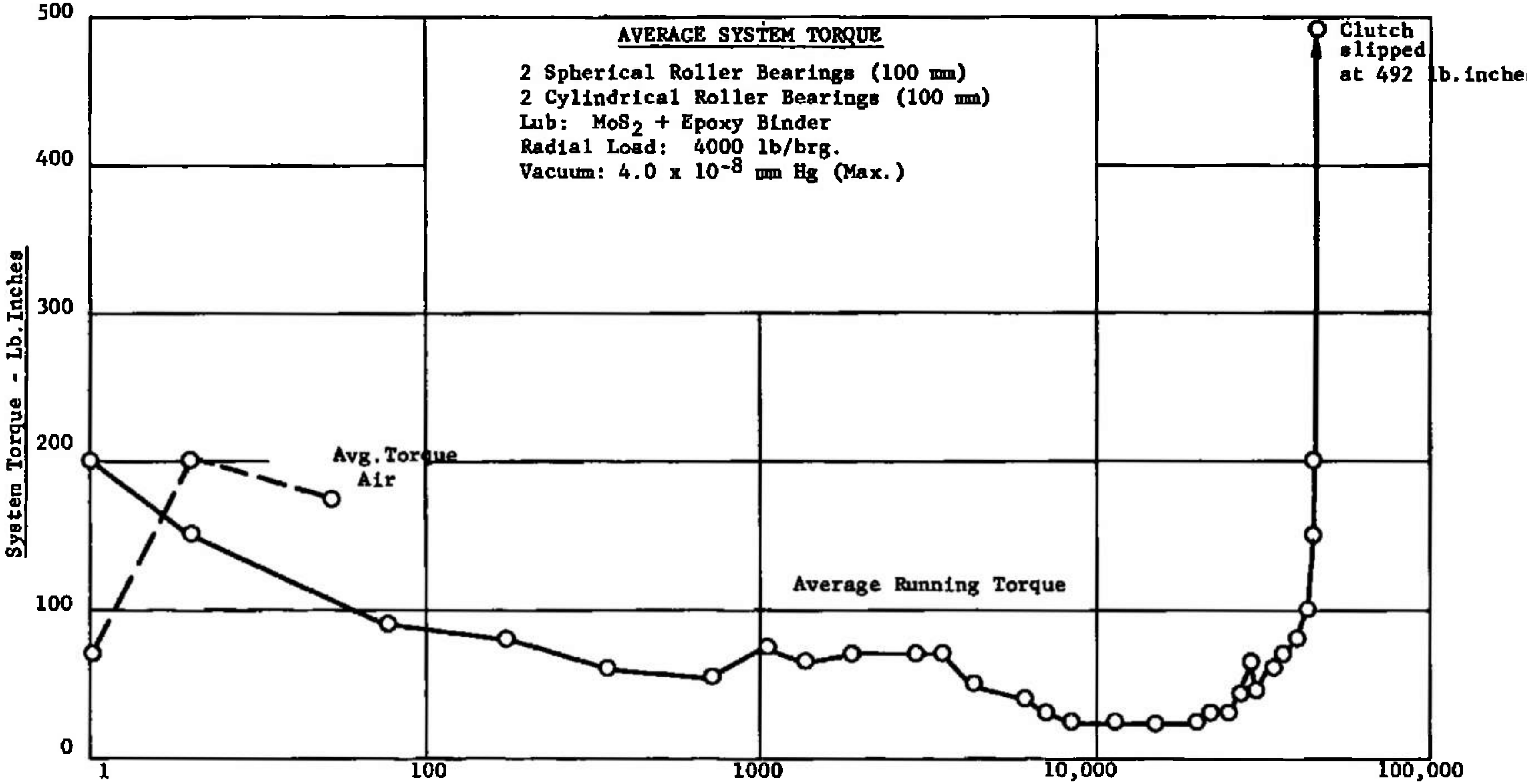

Fig. 44 Average System Torque Curves

Test 7

3. The MoS_2-glass developmental coating indicated considerable promise, although it did not perform as well on the spherical type bearings as it did on the cylindrical type. Improving the method of coating processing should further improve performance on both types of bearings.

4. The longest period of operation with a solid film was obtained with a cylindrical type of roller bearing system, using an MoS_2 + graphite + silicate binder lubricant. The second best performer was a solid film of MoS_2 + epoxy binder on a bearing system consisting of spherical and cylindrical roller bearings.

5. Bearings coated with MoS_2, graphite (with appropriate binder) type of solid films appear to be less sensitive to bearing configuration and operating conditions associated with torque rise, as compared to the thin film plating type of solid film lubricant.

   In some cases, the MoS_2, graphite solid films would wear to a point of marginality before bearing torque increased substantially. The thin platings, on the other hand, needed only a small amount of coating transfer (with original coating still intact) to create a rapid rise in bearing torque. Sufficient drive power to the bearings to allow for redistribution of moderate amounts of pickup would appear to be a factor in promoting further bearing operation. Bearing surface examination indicated lubricant redistribution had taken place in some tests, in others it had not. Metallurgically "worked" small islands of gold or silver on the bearing surfaces resulting from wear debris buildup appeared to be a contributing factor to rolling resistance and subsequent increase in bearing torque in these tests.

6. The Apiezon L, low vapor pressure petroleum distillate grease, tested for comparison purposes with the solid film lubricants, rendered excellent performance results when run for an 80 hour period in a 220A ball bearing. A specially designed Buna N-1010 steel snap shield was used to retain the grease in the bearing.

7. The Buna N-1010 metal bearing shield[*] used in the ball bearing tests would appear to be a useful development from the standpoint of:

   a. Providing a good seal for retention of a lubricant (grease) in a bearing (minimize creepage).

   b. Provide a simple and economic means of re-lubrication by snapping out shield, re-lubricating, and snapping shield back in place.

(*) The shield mold and shields used in the 100 mm ball bearing tests were made by Wasley Products, Plainville, Conn., for the General Electric Company

C. GEAR TEST RESULTS

1. Procedure

Bearings were pressed on shaft and gears assembled in four-square housing. (See Figure 45 for schematic of Four-Square Gear Tester). Axial movement of gears was controlled by the use of shims placed in back of the outer race retainer. Each gear was adjusted so that no axial movement was observed when rotational torque was at its minimum.

Backlash was measured for each set of gear combinations; that is, the forward box and after box assembly was measured separately. The system torque was measured while unloaded and under pre-load conditions by a spring scale and known radius of turning shaft.

Loading was accomplished by means of torsion rod-shaft coupling device with strain gages appropriately mounted and calibrated to indicate the torque applied and subsequent load on the test gears. Strain gage measurements were taken on a Baldwin SR4 Strain Gage Indicator.

Backlash and torque measurements were made before and after each test and when test load changes were made.

A summary of the gear test results for six tests can be seen in Tables 42,43 and 44 . Photographs of the lubricated gear tests can be seen in Figures 46-48

2. Rotary Seal - Gear Tester

Rotary seals were furnished by ARO for the bearing and gear testers.

It was anticipated to use a GE Thymotrol 1 hp motor to drive the gear tester. In addition, motor speed, armature volts and armature current were going to be recorded to give any pertinent information as to increase or decrease in running torque of the Four-Square Tester (power changes).

The rotary feed-through seal drive, when rotated, was measured to be 65 pound inches to rotate, far in excess of the capacity of the thymotrol motor.

Frictional forces were then reduced somewhat to approximately 24 pound inches to allow rotation of the rotary shaft seal. The motor was again tried; however, after running-in for 1 hour, motor current became excessive due to shaft of seal binding. Upon disassembly of rotary seal, considerable wear and brass chips were observed. The seal parts were re-worked and a gear box reducer of 150:1 ratio, and a 1750 rpm fractional-horsepower motor were aligned to drive the seal shaft and Four-Square Gear Tester. This setup performed satisfactorily and because of the limited test schedule, no further attempt was made to monitor power changes during the gear runs.

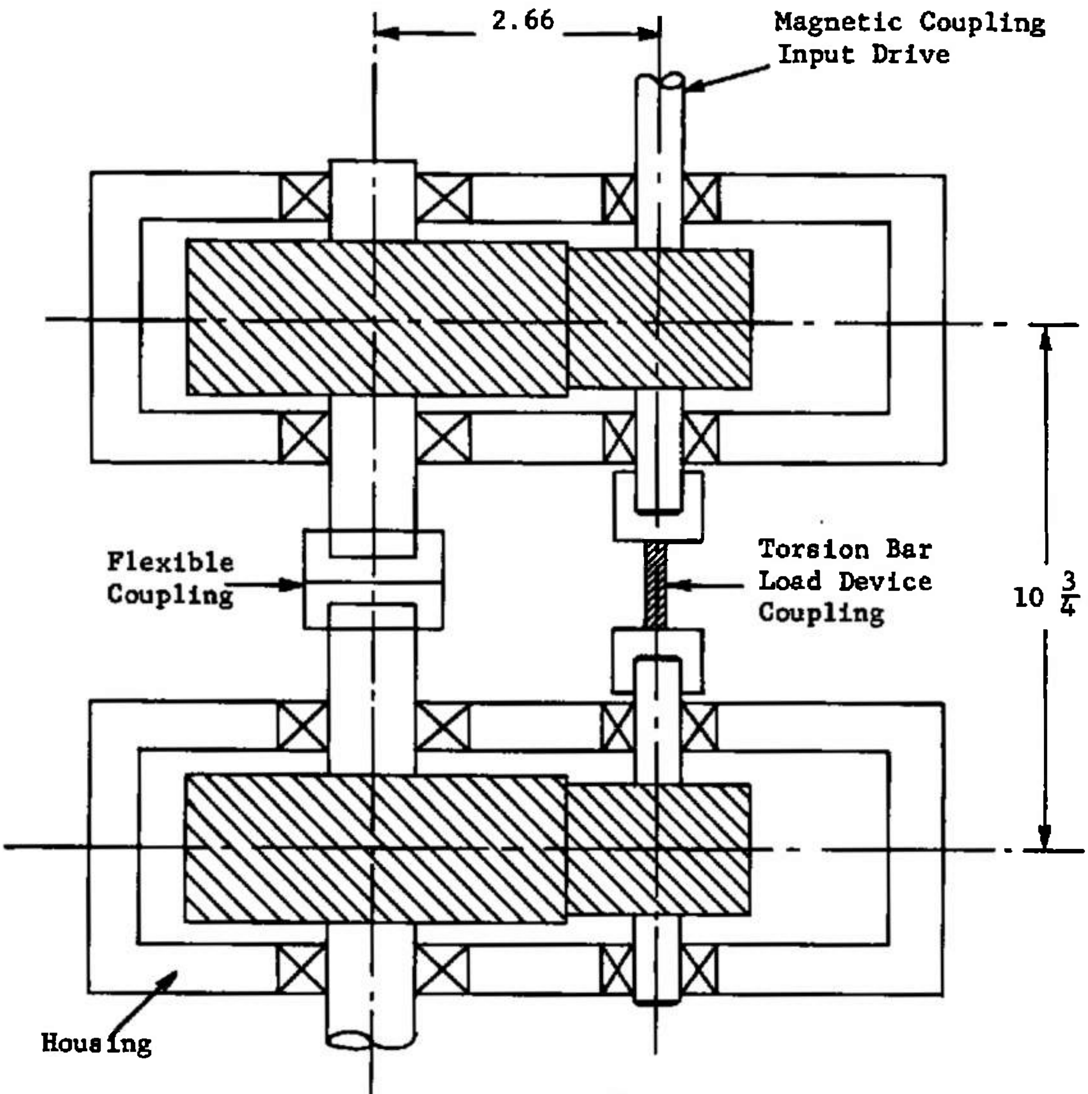

Fig. 45 Schematic of Four-Square Gear Tester

(See also photograph (Figure 28) in Section IX)

TABLE 42

ATL LARGE SCALE GEAR TESTS

TEST NO.	GEAR TYPE	NO. OF GEARS	TANGENTIAL TOOTH LOAD (lbs)	SURFACE COMPRESSIVE STRESS (psi)	TORQUE ON SHAFT (lb-in)	APPROX. TEST DURATION IN VAC. (Hrs.)	TOTAL CYCLES	LUBRICANT	SPEED RPM
1-2	Gears Helical gears 3.2:1 ratio	4	128	47,900	80	71.2	52,140	MoS_2 + Epoxy Binder	11.7
3-4	Helical gears 3.2:1 ratio	4	128 256	47,900 67,700	80 160	136.6 108.0 244.6	95,712 75,816 171,528	23 Kt Gold vs. Silver	11.7
5-6	Helical gears 3.2:1	4	128 256	47,900 67,700	80 160	117.3 139.8 257.1	82,932 98,697 181,629	MoS_2 + Graphite + Silicate Binder	11.7

Gear Materials:

Helical shaft pinion - 1-1/4" P.D., AISI 4140 Steel; Rockwell C 37-39.
Helical (large) - 4" P.D., AISI 4340 Steel; Rockwell C 40-42

Environment:

Gear tests were performed in vacuum between 1.0×10^{-7} mm Hg to 1.0×10^{-8} mm Hg.

TABLE 43

LARGE SCALE GEAR TEST RESULTS (Torque Measurement)

Test No.	Test Period (Hrs.)	Gear Coating Combination 1.25"PD Pinion	Gear Coating Combination 4.0"PD Gear	System[(1)] Torque at No Load lb-in Breakaway	System[(1)] Torque at No Load lb-in Running	System Torque at Preload lb-in Breakaway	System Torque at Preload lb-in Running	Backlash (mils) Top Gear Set	Backlash (mils) Bottom Gear Set	Comments
1	Initial	MoS_2 + Epoxy Binder	MoS_2 + Epoxy Binder	0.8	0.5	4.0 at 80 lb-in pre-load	3.5 at 80 lb-in pre-load	5.1		
	After 71.2 hrs	"	"	4.0	4.0	7.0	3.0	8.1		Approx. 60% of coating intact over gear tooth contact area. Remaining 40% of area coating marginal to depleted exposing base metal in some places.
2	Initial	MoS_2 + Epoxy Binder	MoS_2 + Epoxy Binder						5.1	
	After 71.2 hrs	"	"						5.1	Approx. 80% of coating intact over gear tooth contact area. Remaining 20% of area coating marginal to depleted exposing base metal in some places.

Footnote:

(1) Torque reported is system torque, that is, the combined torques of tests 1 and 2, which are run in the Four Square Tester simultaneously. System torque includes both sets of test gears and support bearings - measurements taken at input shaft.

*All tests were run in vacuum ranging from approximately 1.0×10^{-7} mm Hg to 1.0×10^{-8} mm Hg.

TABLE 43 (Continued)

Test No.	Test Period (Hrs.)	Gear Coating Combination: 1.25"PD Pinion	Gear Coating Combination: 4.0"PD	System Torque at No Load lb-in: Breakaway	System Torque at No Load lb-in: Running	System Torque at Preload lb-in: Breakaway 80#in	System Torque at Preload lb-in: Breakaway 160#in	System Torque at Preload lb-in: Running 80#in	System Torque at Preload lb-in: Running 160#in	Backlash (mils): Top Gear Set	Backlash (mils): Bottom Gear Set	Comments
3	Initial	23Kt Au	Ag	0.8	0.6	7.5		4.0		6.6		
	After 136.5 hrs.			-	-	4.0		2.0		9.1		Coatings appear intact on gears - test suspended to replace failed (cage failure) support bearing.
	Increased load			0.5	0.5		24.0		20.0	6.8		Test continued at higher gear load.
	After 244.6 hrs.			0.3	0.3		24.0		24.0	8.8		Major portion of coating intact - some areas polished - moderate pitting over small tooth area.
4	Initial	Ag	23Kt Au								6.1	
	After 136.5 hrs.										7.4	Coatings on gears appear intact.
	Increased load										3.2	Test continued at higher gear load.
	After 244.6 hrs.										7.6	Approx. 50% of coating still effective - other 50% of gear teeth scored and galled.

TABLE 43 (Continued)

Test No.	Test Period (Hrs.)	Gear Coating Combination		System Torque at No Load lb-in		System Torque at Preload lb-in				Backlash (mils)		Comments
		1.25"PD	4.0"PD	Breakaway	Running	Breakaway		Running		Top Gear Set	Bottom Gear Set	
						80#in	160#in	80#in	160#in			
5	Initial	Graphite + MoS_2 + Silicate Binder	Graphite + MoS_2 + Silicate Binder	0.3	0.3	4.5		4.5		2.2		
	117.3			0.2	0.2	5.0		5.0				Coating intact and polished - excellent condition (both gears)
	Increased load			0.2	0.2		10.0		8.0			
	After 257.1 hr.			0.4	0.4		7.0		-	10.4		Major portion of coating intact and polished - small area of exposed metal on tooth face adjacent to gear face (both gears)
6	Initial	Graphite + MoS_2 + Silicate Binder	Graphite + MoS_2 + Silicate Binder								1.4	
	117.3											
	Increased load											
	After 257.1 hrs.										9.2	Major portion of coating intact and polished - small area of exposed metal on tooth face adjacent to gear face (both gears)

TABLE 44

ATL GEAR TEST LOADS

Torque on Shaft (in.lbs.)	Tangential Tooth Load (lbs.)	"K" Factor	Surface Compressive Stress
10	16	16.8	16,900
20	32	33.6	24,000
40	64	67.2	33,900
80	128	134.4	47,900
160	256	269	67,700
320	512	538	95,800
640	1,024	1,075	136,000
1280	2,048	2,150	192,000
2560	4,096	4,301	270,000

Note: Based on test gears (ATL)

No. teeth pinion - 15 gear - 48

Transverse pitch - 12

Helix angle - 22-1/2^{o} pressure angle - 20^{o}

"K" is a dimensional factor relating gear, pinion geometry, and load, to the maximum compressive stress where:

$$"K" = \frac{W}{Fd}\left(\frac{m_G + 1}{m_G}\right)$$

W = Tangential driving load
F = Active face width
d = Pitch diameter of pinion
m_G = $\frac{\text{number of teeth in gear}}{\text{number of teeth in pinion}}$

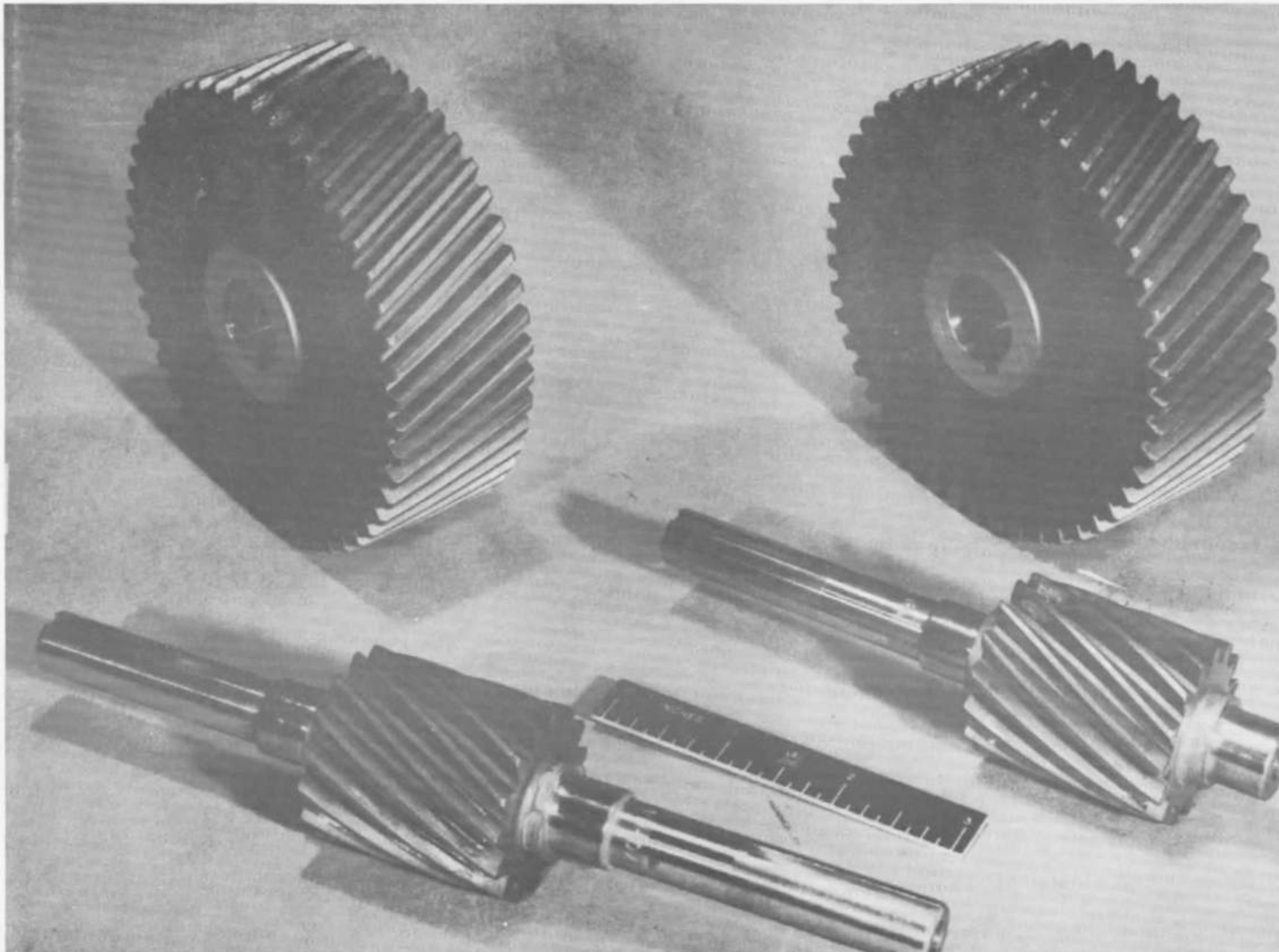

Fig. 46 Gear Tests – Nos. 1 and 2 – the light areas on the tooth surfaces adjacent to the gear faces indicate depletion of lubricant (Lub: MoS_2 + Epoxy Binder).

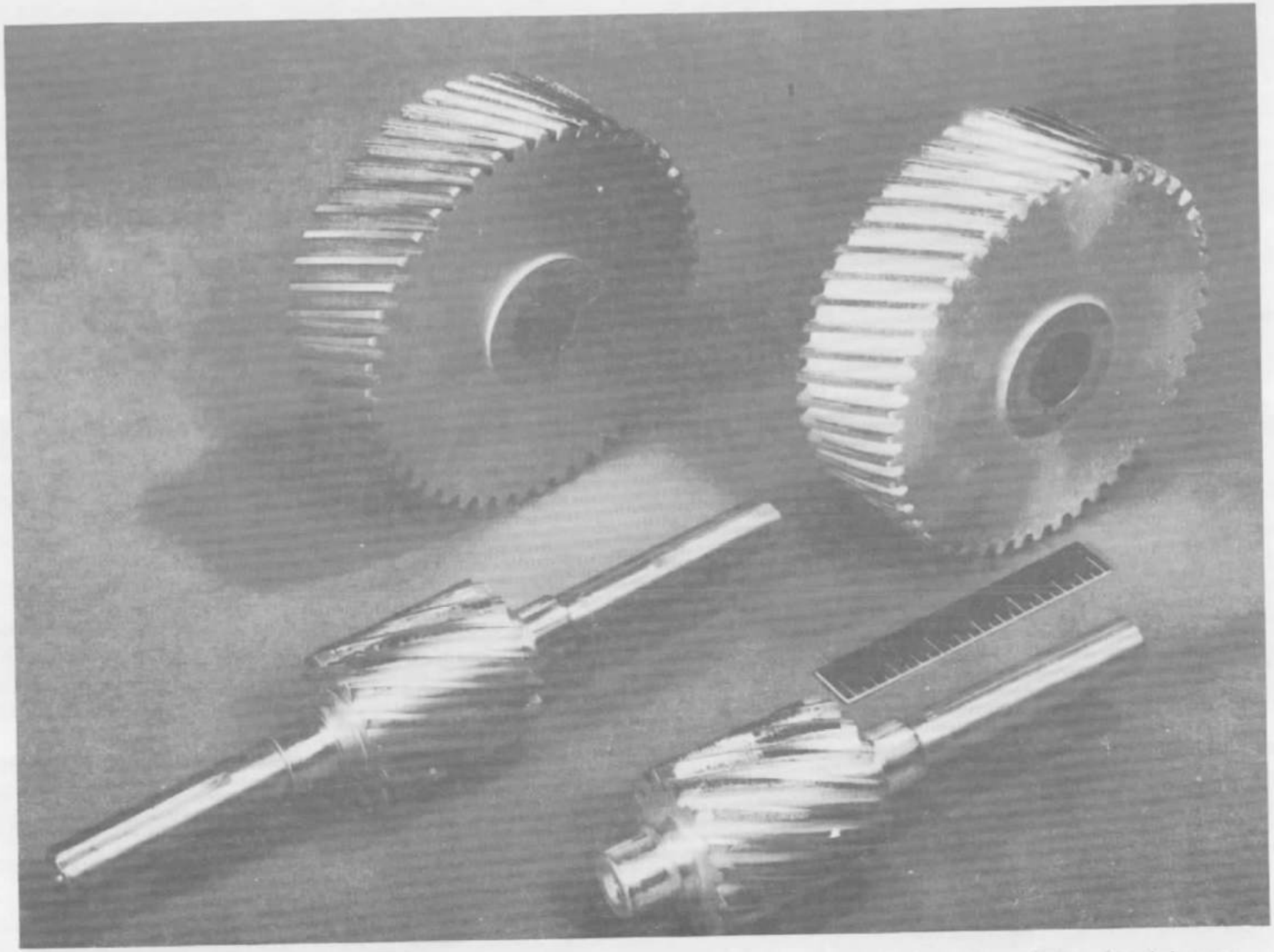

Fig. 47 Gear Tests – Nos. 3 and 4 – Au pinion vs. Ag 4" PD Gear were in relatively good condition, Ag pinion vs. Au 4" PD set experienced considerable scoring and galling.

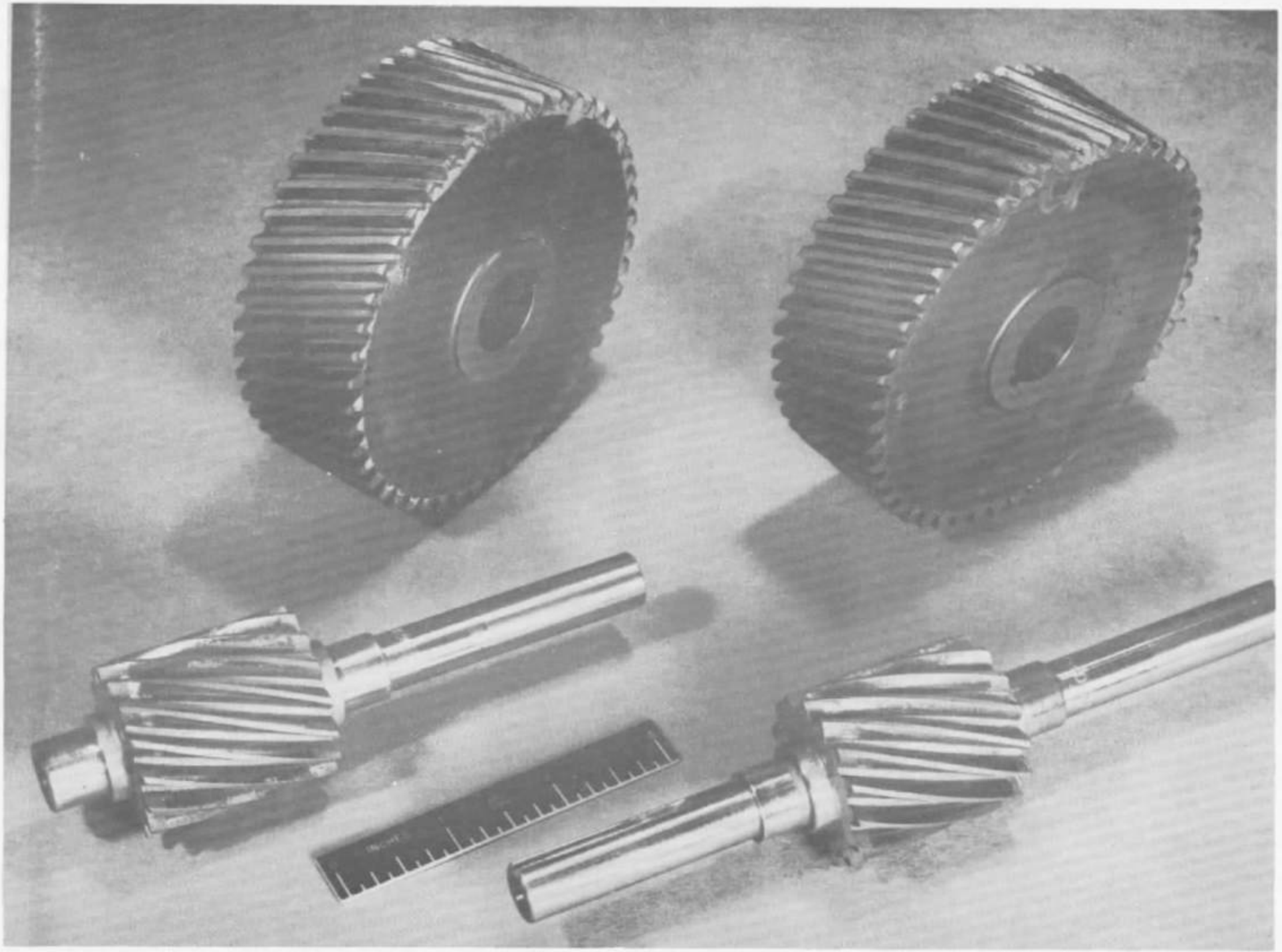

Fig. 48 Gear Tests – Nos. 5 and 6 – both sets of tested gears in relatively good condition – black areas indicate polishing of the MoS_2 + Graphite + Silicate lubricant.

UNCLASSIFIED
Security Classification

DOCUMENT CONTROL DATA - R&D

(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

1. ORIGINATING ACTIVITY (Corporate author) Advanced Technology Laboratories, General Electric Company, Schenectady, N. Y.	2a. REPORT SECURITY CLASSIFICATION UNCLASSIFIED
	2b GROUP NONE

3. REPORT TITLE LUBRICATION OF HEAVILY LOADED, LOW VELOCITY BEARINGS AND GEARS OPERATING IN AEROSPACE ENVIRONMENTAL FACILITIES

This document has been approved for public release its distribution is unlimited. Per DDC TAB 75-15 AD A011 700 Dtd July 1975

4. DESCRIPTIVE NOTES (Type of report and inclusive dates) NONE

5. AUTHOR(S) (Last name, first name, initial) Lee, Ralph E., Jr.

6 REPORT DATE	7a. TOTAL NO. OF PAGES	7b. NO. OF REFS
January 1965	209	55

8a. CONTRACT OR GRANT NO. AF40(600)-1013	9a. ORIGINATOR'S REPORT NUMBER(S) AEDC-TR-65-19
b. PROJECT NO. 7778	
c. Program Area 850E	9b. OTHER REPORT NO(S) (Any other numbers that may be assigned this report) NONE
d. Task 777800	

10. AVAILABILITY/LIMITATION NOTICES

NONE

11. SUPPLEMENTARY NOTES NONE	12. SPONSORING MILITARY ACTIVITY Arnold Engineering Development Center, Arnold AF Station, Tenn.

13. ABSTRACT This report presents the results of a balanced study of bearings and gears for heavily loaded, low velocity space simulator applications, integrated with a research effort directed toward the development of materials, lubricants, application processes, and evaluation and testing techniques. Solid film lubricants consisting of molybdenum disulfide and graphite with silicate and epoxy type binders, and thin film platings of gold and silver, were among the better performers. Molybdenum disulfide-glass, and graphite-aluminum phosphate were among the more successful solid film lubricants developed. Apiezon L, a low vapor pressure petroleum distillate used for comparison purposes exhibited good performance characteristics. Lubricated cylindrical, spherical, ball, and tapered folling element bearings of 30 mm, 50 mm, and 100 mm bore sizes were tested in this effort. The gears tested were helical.

DD FORM 1 JAN 64 1473

UNCLASSIFIED
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14. KEY WORDS	LINK A		LINK B		LINK C	
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Gears						
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